HYDRAULIC TRANSIENTS IN PUMPING MAINS DUE TO POWER FAILURE

A Thesis Submitted
In Partial Fulfilment of the Requirements
for the Degree of

MASTER OF TECHNOLOGY

by
DEVENDRA KUMAR SINGH

to the

DEPARTMENT OF CIVIL ENGINEERING

INDIAN INSTITUTE OF TECHNOLOGY, KANPUR

APRIL, 1987

TO

MY PARENTS

CERTIFICATE

This is to certify that the thesis entitled '' HYDRAULIC TRANSIENTS IN PUMPING MAINS DUE TO POWER FAILURE'' submitted by Shri Devendra Kumar Singh in partial fulfilment of the requirements for the degree of Master of Technology at Indian Institute of Technology, Kanpur is a record of bonafied research work carried out by him under our supervision and guidance. The work embod in this thesis has not been submitted elsewhere for a deg

S. Surya Rao

(Dr. S. Surya Rao)

(Dr. S. Ramaseshan)

Professor
Dept. of Civil Engineering
Indian Institute of Technology, Kanpur

ACKNOWLEDGEMENTS

It is with a deep sense of gratitude, I acknowledge the guidance provided by Prof. S. Surya Rao. I shall remain indebted to him for his kind co-operation, potential outlook and encouragement throughout the tenure of work. Also, I am very happy to acknowledge my gratefulness to Prof. S. Ramaseshan for his valuable and timely help on many occasions in the development of present work.

I would like to extend my thanks to my wife, brothers and sisters for the time that should have been spent with them what was required for the completion of my work.

Thanks are due to my friends, A.K. Dixit, Dayal Singh, Jaykumar, R.K. Pachauri, R.L. Rana, S.K. Shukla and other friends for making my stay at I.I.T. Kanpur enjoyable and memorable one.

I should also acknowledge Mr. Satya Narayan Pradhan for his neat typing of the manuscript.

DKSingh
(D.K. Singh)

CONTENTS

LIST OF TABLES

LIST OF FIGURES

NOMENCLATURE

A	area of pipe
AC	horizontal cross-sectional area of the chamber
a	speed of pressure pulse
C_D	orifice discharge coefficient
CO	initial volume of air inside the chamber
C_{orf}	coefficient of orifice loss
C_v	coefficient of head loss in the valve
D	pipe diameter; characteristic dimension of turbomachine
f	Darcy- Weisbach friction factor
g	gravitational acceleration
H_b	barometric pressure head
H_o	steady-state or mean pressure head
H_p	piezometric head at unknown computational point
H^*_{Pair}	absolute pressure head inside chanber at the end of time step
H_{pv}	head loss in the discharge valve
H_R	rated pressure head of turbomachine
H_{SUC}	height of the liquid surface in the suction reservoir above datum
h	dimensionless pressure head H/H_o or H/H_R
I	polar moment of inertia of rotating parts
i	denotes section number along a pipe
L	pipe length
m	exponent in the polystropic gas equation
n_p	number of parallel pumps
N	rotational speed; the number of reaches into which a pipe line is subdivided for computations
N_R	rated speed of turbomachine
Q	instantaneous discharge at a point
Q_{orf}	discharge through orifice of air chamber

Q_R	rated discharge of turbomachine
T	instantaneous torque on pump or turbine
T_R	rated torque of turbomachine
t	time
V	instantaneous velocity
V^*_{Pair}	volume of air enclosed at the end of time step
x	distance along pipe from left end
z	height of the liquid surface in the chamber above datum at the beginning of time step
z_P	height of the liquid surface in the chamber at the end of time step
α	dimensionless speed ratio
β	dimensionless torque ratio T/T_o or T/T_R
γ	specific weight of water
λ	moment of inertia in terms of the weight of rotating parts of the motor
τ	dimensionless number describing the discharge coefficient and area of opening at a valve
ω	angular velocity.

ABSTRACT

The Tracy Pumping Plant consists of six large units with each pair of pumps connected through a wye branch to a 15 ft. diameter, reinforced concrete pipe which extends approximately 1 mile to the upper canal. Each pump is driven by 22,500-hp motor and delivers water at the rate of 767 cu-ft per sec. to the canal under a rate head of 197 ft. On the discharge side of each pump there is a butterfly valve which closes at a variable rate under the action of servomotor when power tripout occurs at the pump.

A comprehensive study of the hydraulic transients for parallel pump system for the Tracy Pumping Plant and for different modifications of the original plant involving a tota of 14 different cases has been carried out. For this a comput program has been coded in FORTRAN-IV and run on Dec- 1090 system at I.I.T. Kanpur. The system has been analysed without a valve, with a butterfly valve at the pump, a non-return valv at the pump, a combination of a non-return valve at the pump and an intermediate non-return valve at mid section of the discharge line. It has also been analysed with some suction length, increasing the length of discharge line, and doubling the moment of inertia. The system has also been analysed with an airchamber with a check valve upstream of the chamber, by

changing the different parameters involved in the computations. All the cases, being analysed under this thesis, for the transient problems, are due to sudden power tripout.

It has been found that maximum head is 1.4 times the rated head when, system has been analysed without valve. The system, analysed with butterfly valve has shown that 9 percent reducation in maximum head is achieved. The transients for non-return valve at the pump are so severe and quick that special precautions will have to be taken. In the case of resistance dominated system, the maximum head rise is equal to steady-state rated head, and column separation may occur with subsequent rejoining and development of high heads. In case of gravity loading,doubling of moment of inertia has decreased the maximum head and increased the minimum head, and so increase in moment of inertia will be benificial. The system studied with air chamber has indicated that important design parameter for the design of airchamber is $\frac{2C0.a}{Q0.L}$, the values of upsurge and downsurge decrease with increase in this constant. The orifice diameter does not seem to have appreciable effect on upsurge and downsurge.

The findings of the present study provide valuable suggestions for effective and rational design of surge controlling equipments.

CHAPTER 1

INTRODUCTION AND REVIEW OF LITERATURE

1.1 Introduction:

Water-hammer problems relating to turbo-machinery are of great practical importance. If the possible operating conditions of hydraulic-turbine and centrifugal-pump installations are compared, it soon becomes apparent that the pumps are subject to much wider and more involved variations than are the turbines, especially during the transient state of starting, stopping, or emergency operation. In turbines the direction of flow and the direction of rotation are always the same, even in case of a breakdown of machine itself or trouble in the penstock and auxiliary equipment. Thus the machine performance always lies in the quadrant of normal turbine operation, and since its hydraulic characteristics are very well-known in this quadrant, it is completely straight-forward to predict the complete performance during any possible transient condition. On the other hand , under similar conditions with a pump installation, the flow can completely reverse direction, as can the rotation. The machine in this case ceases to be a pump, and after passing through a zone of energy dissipation , becomes a runaway turbine. This great variation in performance gives rise to many questions, such as the runaway speed of the machine as a turbine, the time of reversal, the magnitude of the accelerating

forces, the effect on the surge cycle in the discharge line maximum and terminal reverse rates of flow, and so on. Unfortunately these questions are very difficult to answer, because, although the hydraulic performance of the machine is well known as long as it is acting as a pump, comparatively little study has ever been made of its performance as an energy dissipator or as a turbine.

Therefore, to explore these little-known regions of performance and to attempt to use the resulting information to answer hydraulic transient problems have their own significance.

As the size of hydraulic-power units has increased, more attention has been given to the design and location of auxiliary equipments for the modern hydroelectric plants such as governors, valves, surgetanks, and airchamber. The characteristics of a pipe system during transients depend upon the details of the pipe system and the cases of transients.

Butterfly valve covers a wide range of engineering conditions and is suited to many hydraulic installations. There is reason to believe that this type of valve, because of its simplicity of design, its reasonable cost, and its inherent flexibility, has by no means reached the limit of its possibilities. The widespread use of butterfly valves for controlling waterflow in large pipe lines proves their fitness and adaptability to this type of service.

Water pumping rising mains usually have one or more non-return valves located along the main in addition to the non-return valve near the pumps. These valves may be referred to as intermediate non-return valves to distinguish them from the valve near the pumps. In many systems, intermediate non-return valves are provided with a feeling that it may help in controlling surge pressures, which may not be true. In pumping mains with surge protection devices such as airchamber, the effect of intermediate non-return valves on the transient pressures is of particular interest.

Two extreme cases of pump loading are considered. First a relatively short line with almost all of the head used to lift liquid to a reservoir and second, for the same steady-state head and discharge, a very long pipe line in which substantially all head is used to overcome pipe line friction. Effect of moment of inertia of rotating parts is studied in two cases. Analysis of system with airchamber has also been carried out under different conditions.

1.2 Review of Literature:

A good amount of work has been carried out on hydraulic transient problems with the considerations of different auxiliary equipments. John Parmakian (5) studied the hydraulic transients with butterfly valve at the Tracy Pumping Plant. According to him butterfly valves with a slow time of closure can be of no value in the control of transient

hydraulic conditions because the maximum rise, the maximum drop, and the reversal of flow in the pump discharge line occur within a short period. He also conducted tests including following conditions, to check the adequacy of the pumps, discharge lines, and pressure control-equipment, and also to check the accuracy of water-hammer analysis:

1. Two pumps supplying water to the same discharge line when a power failure occurred at both pump motors.

2. Two pumps supplying water to the same discharge line when a power failure occurred at only one pump motor.

3. One pump supplying water to the discharge line when a power failure occurred at the pump motor.

Streeter (12) gave detailed information about different methods of analysing water-hammer problems, and he included the effects of type of pump loading on transient pressures, valve stroking applied to the pump pressures,etc.

A number of studies (6) have been reported in literature on the effect of non-return valves and control valves (with set closure pattern and time) on hydraulic transients. Murty, Shridharan and Prasad (6) studied the effect of intermediate non-return valves on hydraulic transients following power failure at two pumping systems and they illustrated the effects of (a) location of the intermediate non-return valve, (b) delay in the valve

closure, (c) bypass size, (d) multiple intermediate non-return- valves, and (e) random delays in the closure of valves. They applied method of characteristics to study the effect and concluded that there is no strong case for providing intermediate non-return valve and where provided, they must be accompained by a bypass.

Chaudhry (2) gave a wide description of different boundary conditions for air-chamber and reduced the equations in such a form that it can be used in computer analysis. Allievi (1) discussed the use of air chambers in pumping systems to control transients generated by power failure to the pumps. Charts to determine the size of an airchamber for a pipeline to keep the maximum and minimum pressures within design limits are reported in the literature (12-3). These charts may be used to determine the approximate size of chamber for a pipe line .

1.3 Objectives of the Study:

The major objectives of the study are:

i) to develop a computer program for the transients in the pumping system due to power failure,

ii) to study the effects of butterfly valves or non-return valves at the pump and intermediate non-return valves on the transients,

iii) to study transient in lift dominating versus resistance dominating pipe systems, and

iv) to study the effect of an airchamber for mitigation of the transients and in this process to study the effect of various design variables relating to air chamber.

1.4 Scope of the Study:

Transient problems are broad and diverse, and include areas such as hydroelectric projects, pumped storage schemes, water-supply systems, nuclear power plants, oil pipelines, and industrial pumping system. In the present study transients in pumping systems have been studied. The transients in the Tracy Pumping Plant have been studied by Parmakian and Streeter (8). In the present study, a computer program has been developed for the analysis of transients in the pumping systems. With this program, the Tracy Pumping Plant has been analysed and the results have been compared with those obtained by Parmakian and Streeter. Subsequently, the effects of butterfly valve and non-return valve at the pump, intermediate non-return valve in the discharge line, comparative study of lift dominating versus resistance dominating systems, the effect of an air chamber to mitigate the transients, etc. have also been studied.

1.5 Significance of the Study:

The work carried out will be useful in understanding transients in pump systems with different types of controls and safety measures, and in designing pumping systems to be safe under various possible transient conditions.

1.6 Organisation of the Report:

The study is reported in the following sequence:

i) The analysis and comparison of results for Tracy Pumping Plant, description of sequence of events after power failure, derivation of differential equations of rotating mass and boundary conditions are presented in Chapter 2.

ii) The analysis of hydraulic transients with modifications in the Tracy Pumping Plant has been done in Chapter 3.

iii) General boundary conditions , method of solution of transients and results for airchamber are presented in Chapter 4.

iv) Summary, conclusions and suggestions for future study are given in Chapter 5.

CHAPTER 2

ANALYSIS OF TRANSIENTS OF THE TRACY PUMPING PLANT

2.1 Sequence of Events Following Power Failure:

In the analysis of piping systems which contain turbomachines it is necessary to take into account the changes occurring in the characteristics of the units as their rotational speeds change. During a pump start-up, the discharge valve is usually kept closed to reduce the electtical load on the pump motor; and as the pump speed reaches the rated speed, the valve is gradually opened. It is the usual practice, in a normal pump-stopping procedure, that the discharge valve is first closed slowly, and then the power supply to the pump motor is switched off. In cases of large pipelines designed for low heads, a bypass to the suction reservoir may be used in starting axial flow units. In the above procedure of starting and stopping pump motor, the speed of the pump remains almost constant during the transients in the piping system. Transients caused by emergency pump operations are usually severe, and the pipelines should be designed to withstand positive and negative pressures caused by these operations. There are many operating conditions at large motor-driven centrifugal pump installations which are capable of producing substantial pressure changes in the discharge line. Of these conditions,one of the most important is rapid deceleration of the pump motors because of a power failure.

When power to a pump fails while it is lifting liquid to a reservoir, the following events take place in the absence of the closing of a discharge valve. The pump speed reduces since pump inertia is usually small compared to that of the liquid in the discharge line. Because the flow and the pumping head at the pump are reduced, negative pressure wave propagates downstream in the discharge line, and positive pressure wave propogates upstream in the suction line. Flow in the discharge line reduces rapidly to zero and then reverses through the pump even though the latter may still be rotating in the normal direction. In this condition (i.e., when there is reverse flow through the pump while it is rotating in the normal direction), the pump is said to be in the zone of '' energy dissipation ''. Because of the reverse flow, the pump slows down rapidly, stops momentarily, and then reverses, i.e. the pump is now operating as a turbine. The pump speed increases in the reverse direction until it reaches the runaway speed. With the increase in the reverse speed, the reverse flow through the pump is reduced due to choking effect, and positive and negative pressure waves are produced in the discharge and suction lines, respectively.

When the load on the pumping system is primarily due to fluid friction, as in the case of a long discharge line, the flow is retarded very slowly and no reversal may

occur at all. The danger to the discharge line in this case is the development of column, separation from low pressures. If the pipeline profile is such that the transient-state hydraulic grade line falls below pipeline at any point, vacuum pressure may occur, and the water column in the pipeline may separate at that point. Excessive pressure will be produced when the two columns later rejoin. During the design stages, the possibility of water-column separation should be investigated, and if necessary remedial measures should be taken.

2.2 The Tracy Pumping Plant:

The Tracy Pumping Plant is located in the central valley project near Tracy, Calif, USA. The general arrangement of the plant is shown in Fig. 2. The plant consists of six large pumping units with each pair of pumps connected through a way branch to a 15-ft.diameter, reinforced concrete pipe which extends approximately 1 mile to the upper canal. Each pump is driven by a 22,500-hp motor and delivers water at the rate of 767 cu-ft. per sec to the canal under a rated head of 197 ft. On the discharge side of each pump there is a butterfly valve which closes at a variable rate under the action of servomotor when power failure occurs at the pump. There are several operating conditions at the Tracy Pumping Plant that can cause substantial pressure changes in the discharge lines. The most serious of these operating conditions is the sudden stoppage of the motors as a result of a power failure.

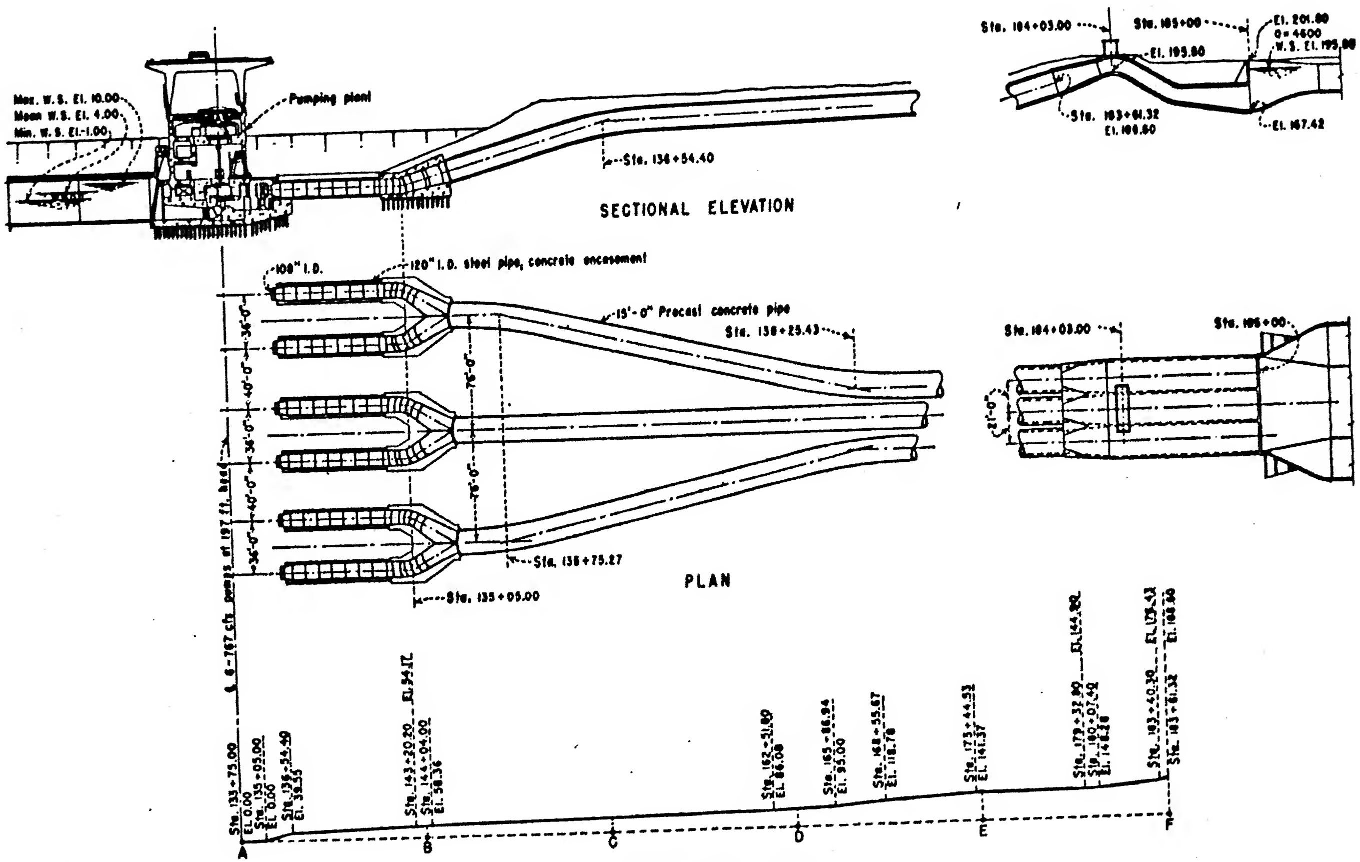

Fig. 2.—Tracy Pumping Plant (California) and Discharge Lines

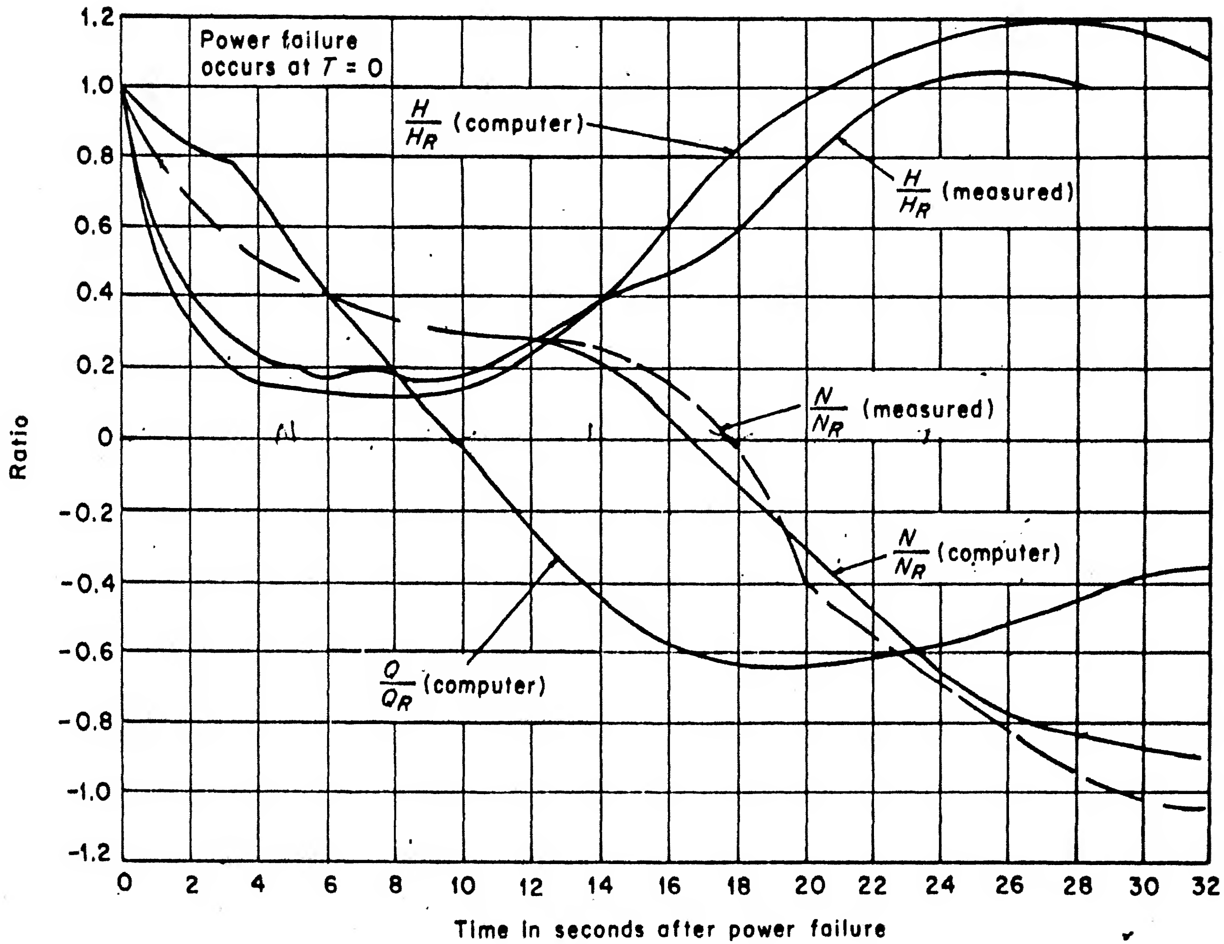

Fig.2a Tracy Pumping Plant failure; results from field tests and from computer analysis. Simultaneous failure of two pumps in parallel discharging to same pipe. H_R *= 107 ft,* Q_R *= 767 cfs,* N_R *= 180 rpm,* L *= 5,130 ft,* D *= 15 ft. Suction pipeline is very short, and is neglected.*

The transients were analysed by Parmakian and Streeter and the time history of the pressure, discharge and speed changes caused by a simultaneous power failure at the motors of two pumps on the same discharge line were presented (Fig. 2a). It can be seen that maximum head rise at the pump is 1.18 H_R. Zone of normal operation is upto 9.8 sec, zone of energy dissipation is from 9.8 sec. to 16 sec., and zone of turbine operation is from 16 sec. The results are shown upto 32 sec. The Tracy Pumping Plant analysed without butterfly valve gave maximum head rise as 275.8 feet as stated by Parmakian.

2.3 Performance Characteristics of a Pump:

The discharge of a centrifugal pump depends upon the rotational speed N, and the pumping head H; and the transient - state speed changes depend upon torque T, and the combined moment of inertia of the pump, motor, and liquid entrained in the pump impeller. Thus, four variables- namely, Q,H,N, and T- have to be specified for the mathematical representation of the dyanamics of a pump. The curve showing the relationships between these variables are called the pump characteristics. There are several methods to store these characteristics for computer analysis. For storing pump characteristics in a digital computer, the method used by Marchal, Flesch, and Suter (12) appears to be most suitable and is used herein.

Although pump-characteristics data in the pumping zone are usually available, only very limited data are generally available for either zone of energy dissipation or the zone of turbine operation. If the complete characteristics data are not available, then the characteristics of a pump having about the same specific speed may be used as an approximation. Homologous theory permits results of model tests to be used; and by using dimensionless, homologous relationships one can store the characteristic curves in the computer in very few units of storage space. To utilize these characteristics, by interpolation for gate position or vane angle, one finds three points on the characteristic curve in the vicinity of the desired conditions. A parabola is fitted through these three points for calculation of boundary conditions at the particular instant under consideration.

Data for prototype pump characteristics are obtained from model test results by using homologous relationships. Two pumps are considered homologous if they are geometrically similar and the stream flow pattern through them is also similar. Then they will satisfy the conditions for dynamic similarity except for viscous effects. The homologous conditions are given by

$$\frac{H}{N^2D^2} = \text{constant}$$

and

$$\frac{ND^3}{Q} = \text{constant} \qquad \text{(Eq.2.1)}$$

In other words, these ratios must be the same for any of the homologous series of units when they are operating in a dynamically similar manner. When studying transient effects in a given turbomachine whose size is fixed and known, diameter of the impeller may be included in the constants; hence

$$\frac{H}{N^2} = \text{constant}$$

$$\frac{Q}{N} = \text{constant} \qquad \text{(Eq. 2.2)}$$

Now if the ratio of H, Q, and N to their rated values H_R, Q_R, N_R are used to yield a dimension less presentation let

$$h = \frac{H}{H_R}, \quad v = \frac{Q}{Q_R}, \quad \text{and} \quad \alpha = \frac{N}{N_R} \qquad \text{(Eq.2a)}$$

The dimensionless-homologous relations are

$$\frac{h}{\alpha^2} = \text{constant}, \quad \frac{v}{\alpha} = \text{constant} \qquad \text{(Eq.2b)}$$

Since α becomes zero while analysing transients for all four zones of operation $\frac{h}{\alpha^2}$ becomes infinite. To avoid this, the parameter $\frac{h}{(\alpha^2+v^2)}$ instead of $\frac{h}{\alpha^2}$ may be used.

The signs of v and α depends upon the zone of operation. In addition to the need to define a different characteristic curve for each zone of operation, $\frac{\alpha}{v}$ becomes infinite for v=0. To avoid this, a new variable θ may be defined as

$$\theta = \tan^{-1} \frac{\alpha}{v} \qquad \text{(Eq.2c)}$$

and then characteristic curve may be plotted between θ and $h/(\alpha^2+v^2)$. By definition, θ is always finite, and its value varies between 0° and 360° for the four zones of operation given in Table 2.1 below:

TABLE 2.1 ZONES OF PUMP OPERATION

Zone of operation	Sign of v	Sign of α	Range of θ
Pump	+	+	$0^\circ \leq \theta \leq 90^\circ$
Energy dissipation	-	+	$90^\circ \leq \theta \leq 180^\circ$
Turbine	-	-	$180^\circ \leq \theta \leq 270^\circ$
Turbine energy dissipation	+	-	$270^\circ \leq \theta \leq 360^\circ$

If T be the shaft torque required for given speed and discharge ratios, the dimensionless-homologous relation for torque ratio, $\beta = \frac{T}{T_R}$ are

$$\frac{\beta}{\alpha^2} = \text{constant}$$

$$\text{and} \quad \frac{v}{\alpha} = \text{constant} \qquad \text{(Eq.2d)}$$

Similar to the head characteristic curve, the torque characteristic curve may be plotted between $\beta/(\alpha^2+v^2)$ and θ.

2.4 Equations of Conditions Imposed by Pump:

The pump characteriestics curve can be used if we have discrete points on these curve at equal interval of θ, between the range $\theta = 0^{\circ}$ and $\theta = 360^{\circ}$, are stored in the computer. Each segment of these curves between points stored in the computer may be approximated by straight lines. If sufficient number of points are stored in the computer then error introduced by approximating these curves by segmental straight lines is negligible.

Given that variables α, v, h and β are known at i-th time step it is needed to calculate these variables at the end of time step denoted respectively by α_p, v_p, h_p, and β_p.

Since small time steps may economically be used in carrying out the transient calculation on a large digital computer, the speed change of a turbomachine may be very closely estimated for a particular time step if one extrapolates the speed and discharge ratios to the mid point of a small time step, calculates the torque for these conditions, and then determines the speed change at the end of the time step from torque and moment of inertia of rotating members. To determine the values of these variables it is in turn necessary to determine the equation of the segment of pump characteristics corresponding to α_p and v_p. As a first estimate, their values are determined by extrapolation from the known values from the previous time step i.e.,

$$\alpha_e = \alpha_i + \frac{\Delta\alpha_{i-1}}{2}$$

(Eq.2.3)

$$v_e = v_i + \frac{\Delta v_{i-1}}{2}$$

in which α_e and v_e are estimated values at the end of i-th time step, α_i and v_i refers to known values at the beginning of i-th time step, and $\Delta\alpha_{i-1}$ and Δv_{i-1} are changes in these variables during the (i-1)th time step. Now, the grid points on either side of $\Theta = \tan^{-1}(\alpha_e/v_e)$ are searched, and the ordinates $h/(\alpha^2+v^2)$ and $\beta/(\alpha^2+v^2)$ for these grid points are determined from the stored values. From these, the constants a_1, a_2, a_3, a_4 for the equation of the segmental straight line are determined. Now, assuming that the points corresponding to α_p, v_p, h_p and β_p lie on these straight lines, then

$$\frac{h_p}{\alpha_p^2 + v_p^2} = a_1 + a_2 \tan^{-1}\frac{\alpha_p}{v_p} \qquad \text{(Eq.2.4)}$$

and

$$\frac{\beta_p}{\alpha_p^2 + v_p^2} = a_3 + a_4 \tan^{-1}\frac{\alpha_p}{v_p} \qquad \text{(Eq.2.5)}$$

in which a_1, a_2 and a_3, a_4 are constants for the straight lines representing, respectively, the head and torque characteristics.

Boundary conditions at the turbomachines may include combinations of units in parallel and in series cases in which some units lose power while others continue

to operate, and action of valving. Since the pipeline water-hammer is conveniently calculated by the method of characteristics, the inclusion of various appurtenances along the pipeline to control the transients may be included or omitted from the program as an option that in no way changes the more complex boundary conditions at the turbo-machines.

Refering to Fig.2.1 the following equations can be written for the total head at the pump:

$$H_{P_{i,1}} = H_{suc} + H_P - H_{Pv}. \qquad \text{(Eq.2.6)}$$

in which,

H_{suc} = height of the liquid surface in the suction reservoir above datum,

H_P = pumping head at the end of time step

H_{Pv} = head loss in the discharge valve.

The value of head loss is given by

$$H_{Pv} = C_v \times Q_{P_{i,1}} \times |Q_{P_{i,1}}| \qquad \text{(Eq.2.7)}$$

in which C_v = coefficient of head loss in the valve, and $Q_{P_{i,1}}$ = discharge at section (i,1).

2.4.1 Characteristic Equation for Discharge Pipe:

As the suction line is short, it may be neglected in the analysis, The negative characteristic equation may

be written along AP, (Fig. 2.2)

$$Q_P = Q_A - \frac{g.A}{a} \times H_A - \frac{f.Dt}{2.D.A} \times Q_A|Q_A| + \frac{g.A}{a} \times H_P$$

or $Q_P = Cn + Ca.\ H_P$ (Eq.2.8a)

where $Cn = Q_A - \frac{g.A}{a} \times H_A - \frac{f.Dt}{2.D.A} \times Q_A|Q_A|$

and $Ca = \frac{g.A}{a}$

Reffering to Fig. 2.1 characteristic equation for section (i,1) may be written as,

$$Q_{P_{i,1}} = Cn + Ca.H_{P_{i,1}} \quad \text{(Eq.2.8)}$$

2.4.2 Continuity Equations:

Since there is no storage between the suction reservoir and section (i,1) it can be written as

$$Q_{P_{i,1}} = Q_P \quad \text{(Eq.2.9)}$$

in which Q_P = flow through the pump at the end of time step.

2.5 Differential Equation of Rotating Masses and Calculation of Speed Change:

When the power to the pump motor is interrupted or suddenly cut-off, the deceleration of the pump at any instant depends on the flywheel effect of the rotating parts

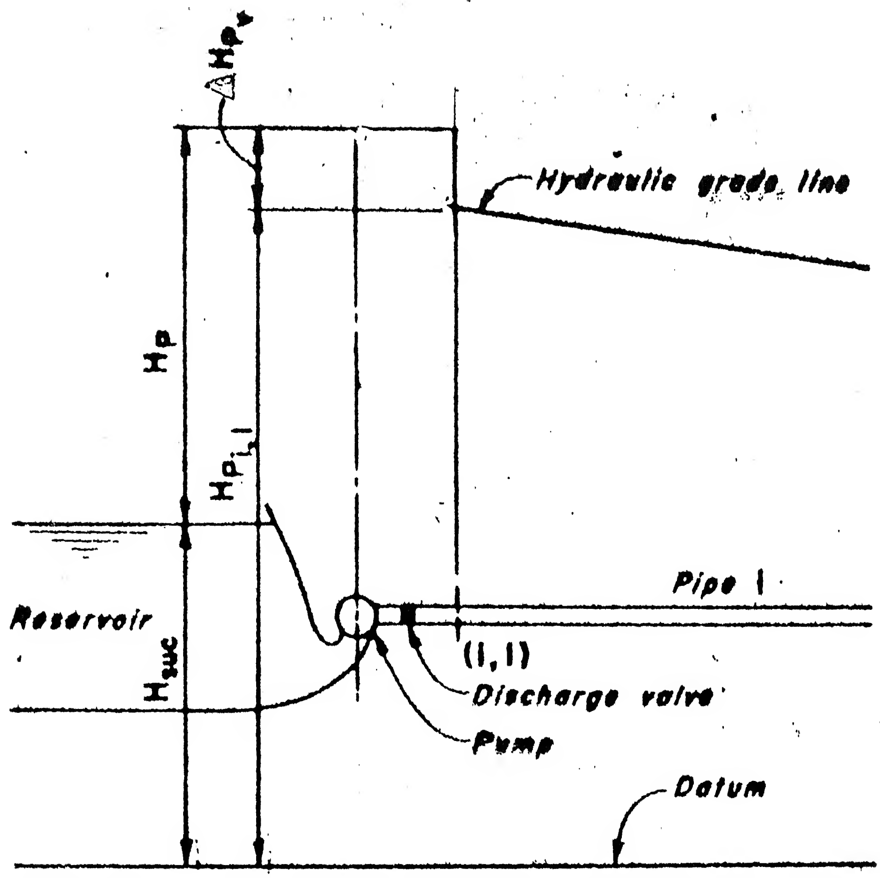

Fig 2.1 Notation of Boundary Condition for Pump

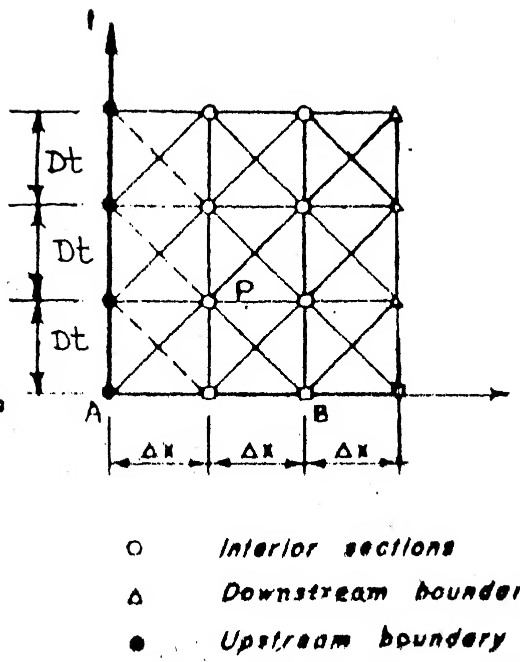

Fig 2.3 Characteristic Grid

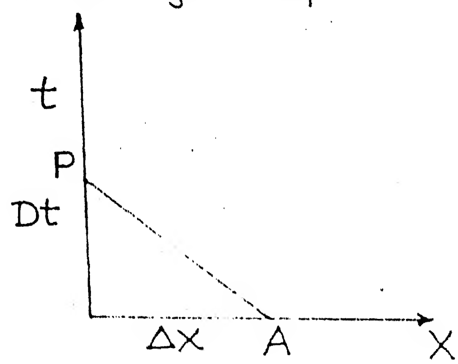

Fig 2.2 Characteristics At Upstream Boundary

of the pump and motor and the instantaneous torque exerted by pump impeller. For a rotating system, the accelerating torque is equal to the product of the mass moment of inertia of the rotating system and the angular acceleration. Following a power failure at the pump motor, the decelerating torque of the rotating system corresponds to the pump-input torque prior to power failure. If the decelerating torque is taken as positive,

$$T = -I \frac{d\omega}{dt} = -\frac{\lambda}{g} \frac{d\omega}{dt} \qquad \text{(Eq.2.10)}$$

in which

T = the pump input torque corresponding to a given speed and head, in pound-feet

I = the mass moment of inertia of the ratating parts, in pound-feet squared

ω = the angular velocity of pump and motor shaft, in radians per second

t = the time in seconds

λ = the flywheel effect, or moment of inertia in terms of the weight of the rotating parts of the motor, pump and entrained water, in pound-feet square

g = the gravitational acceleration, in feet per second squared.

Because the pump torque varies with the speed, Eq.2.10 is applicable for specific instants of time only.

For a small time interval ($\Delta t = t_2 - t_1$), Eq.2.10 can be approximated as

$$\frac{T_1+T_2}{2} = \frac{\lambda}{g}\ \frac{\omega_2-\omega_1}{\Delta t} = \frac{2\pi\lambda}{60.g} \times \frac{N_1-N_2}{\Delta t} \qquad \text{(Eq.2.11)}$$

in which N is the pump speed, in revolutions per minute. Eq.2.11 can be written as

$$\alpha_1-\alpha_2 = \frac{15.gT_R}{\pi\lambda N_R}\ (\beta_1+\beta_2)\ \Delta t \qquad \text{(Eq.2.12)}$$

in which $\alpha = N/N_R$, $\beta = T/T_R$ and the subscript R is used to denote the rated conditions. :declerating torque at the rated head and speed of the pump is

$$T_R = \frac{60\gamma H_R Q_R}{2.\pi.N_R\eta_R} \qquad \text{(Eq.2.13)}$$

in which γ is the specific weight of water, in pounds per cubic ft. ; H represent the pressure head for surge conditions measured above the pump-intake; watersurface elevation, in feet; Q is the pump discharge, in cubic feet per second and η denotes the pump efficiency. Then

$$\alpha_1 - \alpha_2 = K_1(\beta_1+\beta_2)\Delta t \qquad \text{(Eq.2.14)}$$

in which $K_1 = \dfrac{450.g.\gamma.H_R Q_R}{\pi^2.\lambda\,\eta_R\, N_R^2} = \dfrac{91,600\ H_R Q_R}{\lambda\,\eta_R\, N_R^2}$ (Eq.2.15)

Eq.2.14 can be rewritten as

$$\alpha_P - C_6.\beta_P = \alpha + C_6.\beta \qquad \text{(Eqn.2.16)}$$

in which $C_6 = \dfrac{-15\ T_R.\Delta t}{\pi.\omega R^2\ N_R}$ (Eq.2.17)

2.6 Boundary Conditions for Parallel Pumps- Short Suction Line:

The continuity equation for parrallel pumps is

$$Q_{P_{i,1}} = n_p . Q_P \qquad \text{(Eq. 2.18)}$$

in which n_p = number of parrallel pumps.

Depending upon the length of the suction line, boundary conditions for parallel pumps may be divided into the following two cases:

i) short suction line

ii) long suction line

In case of short suction line, the water-hammer waves are neglected. By eliminating $H_{P_{i,1}}$, H_{Pv} and $Q_{P_{i,1}}$ from eqns. 2.6, 2.7, 2.8, and 2.9 and by using Q_R and H_R as reference values, the resulting equation may be written as

$$Q_R . v_P = Cn + Ca . H_{Suc} + Ca . H_R . h_P - Ca . C_v . Q_R^2 \, v_P |v_P| \qquad \text{(Eq. 2.19)}$$

Now we have four eqns.--i.e. Eq. 2.4, 2.5, 2.26, and 2.19 in four unknowns $\alpha_P, h_P, v_P, \beta_P$.

By substituting for h_P from eqns. 2.4 into 2.19 and for β_P from eqns. 2.5 into eqn. 2.16 and simplyfying, we have

$$F_1 = Ca . H_R . a_1 (\alpha_P^2 + v_P^2) + Ca . H_R \, a_2 (\alpha_P^2 + v_P^2) \tan^{-1} \frac{\alpha_P}{v_P}$$

$$- Q_R . v_P . n_P - Ca . C_v . Q_R^2 \, v_P |v_P| + Cn + Ca . H_{suc} = 0 \qquad \text{(eq. 2.20)}$$

$$F_2 = \alpha_P - C_6 . a_3(\alpha_P^2 + v_P^2) - C_6 . a_4(\alpha_P^2 + v_P^2) \tan^{-1} \frac{\alpha_P}{v_P} - \alpha - C_6 . \beta = 0$$

(Eq.2.21)

$$\therefore \frac{\partial F_1}{\partial v_P} = Ca.H_R(2.a_1.v_p - a_2.\alpha_P + 2.a_2.v_P \tan^{-1} \frac{\alpha_P}{v_P}) - n_P . Q_R$$

$$- 2.Ca.C_v . Q_R^2 |v_P|$$

(Eq.2.22)

$$\frac{\partial F_1}{\partial \alpha_P} = Ca.H_R(2a_1\alpha_P + a_2.v_P + 2.a_2.\alpha_P \tan^{-1} \frac{\alpha_1}{v_P})$$

(Eq.2.23)

$$\frac{\partial F_2}{\partial \alpha_P} = 1 - C_6(2.a_3.\alpha_P + a_4.v_P + 2.a_4.\alpha_P \tan^{-1} \frac{\alpha_P}{v_P})$$

(Eq.2.24)

and

$$\frac{\partial F_2}{\partial v_P} = C_6(-2.a_3.v_P + a_4.\alpha_P - 2.a_4.v_P \tan^{-1} \frac{\alpha_P}{v_P})$$

(Eq.2.25)

Let $\alpha_P^{(1)}$ and $v_P^{(1)}$ be the initially estimated values of solution, which may be taken equal to α_e and v_e as determined from Eq. Then a better solution of Eqs.2.20 and 2.21 is

$$\alpha_P^{(2)} = \alpha_P^{(1)} + \delta\alpha_P/2$$

(Eq.2.26)

$$v_P^{(2)} = v_P^{(1)} + \delta v_P/2$$

(Eq.2.27)

subscript (1) indicates estimated values and subscript (2) indicates valves after first iteration.

in which

$$\delta\alpha_P = \frac{F_2 . \frac{\partial F_1}{\partial v_p} - F_1 . \frac{\partial F_2}{\partial v_P}}{\frac{\partial F_1}{\partial \alpha_P} . \frac{\partial F_2}{\partial v_P} - \frac{\partial F_1}{\partial v_P} . \frac{\partial F_2}{\partial \alpha_P}}$$

(Eq.2.28)

and $$\delta v_p = \frac{F_2 \cdot \frac{\partial F_1}{\partial \alpha_p} - F_1 \cdot \frac{\partial F_2}{\partial \alpha_p}}{\frac{\partial F_1}{\partial v_p} \cdot \frac{\partial F_2}{\partial \alpha_p} - \frac{\partial F_1}{\partial \alpha_p} \cdot \frac{\partial F_2}{\partial v_p}} \qquad \text{(Eq.2.29)}$$

2.7 Procedure for Solving the Transients:

The general procedure for solving the transient problem is to first set up the steady-state case for the given elevations of reservoirs, and pump and pipeline characteristics, computing steady-state head H_o, steady-state discharge Q_o, and the velocity, and elevation of hydraulic grade line at equally-spaced sections along the suction and discharge pipes. The basic data required for analysis of the problem are of Tracy Pumping Plant given in Table 2.2 and the head and torque data requiredare given in Table 2.3. The tabular data of Table 2.3 are (α/v or v/α) vs($\frac{h}{\alpha^2}$ or $\frac{h}{v^2}$) and ($\frac{\alpha}{v}$ or $\frac{v}{\alpha}$) vs($\frac{\beta}{\alpha^2}$or $\frac{\beta}{v^2}$). To use Table 2.3 in the computer it is converted to Table .2.4 in which ($\frac{\alpha}{v}$ or $\frac{v}{\alpha}$) vs ($\frac{h}{\alpha^2+v^2}$) and ($\frac{\alpha}{v}$ or $\frac{v}{\alpha}$) vs ($\frac{\beta}{\alpha^2+v^2}$) values are stored. By this computation ,theta ($\theta = \tan^{-1}\alpha/v$) values were found to be at unequal intervals. So this table is further changed to Table 2.5 to obtain the values of θ at equal intervals . After obtaining Table 2.5 , the speed change is calculated from Eq.2.3 Section 2.4.

A sketch of sectional elevation and plan of pump and discharge line is given in Fig. 2. It is clear from the figure that the upstream boundary condition is imposed by pump characteristics and dounstream boundary condition is

TABLE 2.2

Basic Data and Results for different cases without Air Chamber

Item	Data & Results							
Case No	1	2	3	4	5a	5b	6a	6b
Rated Pumping Head, in feet Suction Reservoir level in feet	197 4	197 4	197 4	197 4	197.0 4	197.0 4	197.0 4	197 4
Length of Suction pipe in feet	0	0	0	0	1282.5	1282.5	7695	7695
Length of discharge pipe in ft	5130	5130	5130	5130	3847.5	3847.5	23085	23085
Dia. of discharge pipe in feet	15	15	15	15	15	15	15	15
ated pump discharge in cubic feet per second	767	767	767	767	767	767	767	767
Water-hammer wave travel time in each discharge pipe in sec.	1.75	1.75	1.75	1.75	1.31	1.31	7.88	7.88
Motor speed in rpm	180	180	180	180	180	180	180	180
RESULTS								
Max pressure head in feet	274.1	259.1	362.7	273.4	272.5	249.9	201.0	201.0
Distance of max pressure point from the pump	0	0	0	2565	0	0	0	0
Minimum pressure head in feet	24.6	29.0	26.9	53.1	32.0	58.6	-21.8	-12.4
Distance of minimum pressure point from the pump	0	0	0	0	0	0	0	0

Table 2.3 Head and torque data for pump of specific speed 1,800 (gpm units)
$(v/\alpha) = (\alpha/v) = 0.1$ for normal and turbine zone and -0.1 for energy-dissipation zone; data are based on rated values of H, Q, N, and T.

$\frac{\alpha}{v}$ or $\frac{v}{\alpha}$	0.	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
HAN	1.288	1.287	1.284	1.272	1.256	1.236	1.206	1.166	1.118	1.061	1.000
HVN	-0.556	-0.472	-0.376	-0.270	-0.167	-0.080	0.108	0.277	0.525	0.740	1.000
HAD	1.288	1.291	1.312	1.347	1.382	1.431	1.500	1.600	1.715	1.844	1.992
HVD	0.692	0.742	0.807	0.883	0.975	1.086	1.212	1.371	1.542	1.755	1.992
HAT	0.634	0.652	0.668	0.684	0.705	0.732	0.764	0.806	0.861	0.927	1.011
HVT	0.692	0.656	0.631	0.631	0.641	0.663	0.700	0.761	0.834	0.910	1.011
BAN	0.450	0.504	0.567	0.633	0.707	0.772	0.833	0.886	0.931	0.969	1.000
BVN	-0.372	-0.292	-0.192	-0.049	0.075	0.215	0.348	0.483	0.645	0.815	1.000
BAD	0.450	0.393	0.372	0.367	0.381	0.419	0.484	0.582	0.700	0.850	1.040
BVD	0.865	0.895	0.915	0.935	0.961	0.978	0.990	0.999	1.015	1.024	1.040
BAT	-0.684	-0.499	-0.332	-0.196	-0.098	-0.042	0.023	0.110	0.200	0.316	0.455
BVT	0.865	0.831	0.800	0.800	0.750	0.700	0.650	0.600	0.558	0.508	0.455

In the notation, H refers to a head ratio, A means division by α^2, D is the zone of energy dissipation, and N is the normal zone. V means division by v^2, T is the zone of turbine operation, B refers to a torque ratio.

Table 2.4 Converted head and torque data for pump of specific speed 1,800 (gpm units)

$\frac{\alpha}{v}$ or $\frac{v}{\alpha}$	0.	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
$\frac{h}{\alpha^2+v^2}$ HAN	1.288	1.274	1.235	1.167	1.083	0.989	0.887	0.783	0.682	0.586	0.500
,, HVN	-0.556	-0.467	-0.361	-0.2[illegible]8	-0.144	-0.064	0.079	0.186	0.320	0.409	0.500
,, HAD	1.288	1.278	1.262	1.236	1.191	1.145	1.103	1.074	1.046	1.019	0.996
,, HVD	0.692	0.735	0.776	0.810	0.841	0.869	0.891	0.920	0.940	0.981	0.996
,, HAT	0.634	0.646	0.642	0.628	0.608	0.586	0.562	0.541	0.525	0.512	0.506
,, HVT	0.692	0.650	0.607	0.580	0.553	0.531	0.515	0.511	0.509	0.503	0.506
$\frac{\beta}{\alpha^2+v^2}$ BAN	0.450	0.500	0.545	0.581	0.610	0.618	0.613	0.595	0.568	0.535	0.500
,, BVN	-0.372	-0.289	-0.185	-0.045	0.065	0.172	0.256	0.324	0.393	0.450	0.500
,, BAD	0.450	0.389	0.358	0.337	0.328	0.335	0.356	0.391	0.427	0.470	0.520
,, BVD	0.865	0.886	0.880	0.858	0.828	0.783	0.728	0.671	0.619	0.566	0.520
,, BAT	-0.684	-0.494	-0.319	-0.180	-0.084	-0.034	0.017	0.074	0.122	0.175	0.228
,, BVT	0.865	0.823	0.769	0.734	0.647	0.560	0.478	0.403	0.340	0.281	0.228

In the notation, h refers to a head ratio, α refers to speed ratio, v refers to discharge ratio, β refers to torque ratio.

Table 2.5 Pump Characteristics Data at Equal Interval of θ.

$\theta = \tan^{-1}\frac{\alpha}{v}$	NS = 1800 (gpm units)				
	$\frac{h}{\alpha^2+v^2}$	$\frac{\beta}{\alpha^2+v^2}$	θ	$\frac{h}{\alpha^2+v^2}$	$\frac{\beta}{\alpha^2+v^2}$
0^o	-0.556	-0.372	140	0.957	0.598
5	-0.478	-0.299	145	0.920	0.671
10	-0.386	-0.209	150	0.886	0.740
15	-0.284	-0.089	155	0.860	0.798
20	-0.181	0.026	160	0.830	0.839
25	-0.090	0.137	165	0.799	0.865
30	-0.076	0.309	170	0.766	0.881
35	-0.186	0.324	175	0.730	0.883
40	0.356	0.416	180	0.692	0.865
45	0.500	0.500	185	0.655	0.828
50	0.643	0.555	190	0.617	0.782
55	0.783	0.595	195	0.589	0.745
60	0.909	0.614	200	0.563	0.678
65	1.020	0.615	205	0.538	0.589
70	1.113	0.600	210	0.519	0.496
75	1.188	0.570	215	0.511	0.403
80	1.244	0.535	220	0.507	0.316
85	1.276	0.494	225	0.506	0.228
90	1.288	0.450	230	0.520	0.143
95	1.279	0.397	235	0.541	0.074
100	1.266	0.365	240	0.567	0.006
105	1.244	0.344	245	0.593	-0.050
110	1.207	0.331	250	0.615	-0.118
115	1.157	0.333	255	0.632	-0.224
120	1.112	0.351	260	0.642	-0.360
125	1.074	0.391	265	0.646	-0.684
130	1.035	0.444	270	0.634	-0.684
135	0.996	0.520			

due to constant head reservoir. In case 1. of the study there is no valve just downstream of the pump and the transient is due to failure of two pump motors.

The suction pipe length is zero in this case and length of discharge line is 5130 ft. The plant consists of six large pumps with each pair of pumps connected through a wye branch to a 15-ft. diameter, reinforced concrete pipe which extends approximately 1 mile to the upper canal. Each pump is driven by a 22, 500-hp motor and delivers water at the rate of 767 cu-ft. per sec. to the canal under a rated head of 197 ft. The transient due to simultaneous falure of both the pumps are studied.

At the upstream boundary we have four Eqs. 2.4, 2.5, 2.16 and 2.19 in four unknowns α_p, v_p, h_p and β_p. These four equations are reduced to two non-linear equations in two unknowns, α_p and v_p. These two Eqs. 2.20 and 2.21 are then solved by Newton-Raphson method, in which a solution of the equation is first guessed, which is then refined to a required degree of accuracy by successive iterations. Having determined α_p and v_p, it is verified whether the segment of the pump characteristic used in the computation corresponds to α_p and v_p. If it does not, then α_e and v_e are assumed equal to α_p and v_p. If the correct segment was used, then h_p and β_p are determined from Eqs. 2.4 and 2.5, H_p and Q_p are determined from the equations $H_p = h_p x H_R$ and $Q_p = v_p x Q_R$; and $H_{p_{i,1}}$ and $Q_{p_{i,1}}$ from

Eqs. 2.6 and 2.9 . The values of α and v are initialised for the next time step (i.e. $\alpha=\alpha_P$ and $\beta=\beta_P$), and the solution progresses to the next time step. To avoid an unlimited number of interations in the case of divergence of solution, a counter may be used so that the computations are stopped if the number of interations exceed a specified value.

For computations at interior points, let conditions at time $t= t_o$ be known, these are initial steady-state conditions at time $t_o=0$, The unknown conditions at $t_o+ Dt$ are to be determined. Referring to Fig.2.3 , positive characteristic equation may be written AP,

$$Q_P= Q_A+ \frac{g.A}{a} . H_A - \frac{f.Dt}{2.D.A} Q_A|Q_A| - \frac{g.A}{a} . H_P$$

$$\text{or} \quad Q_P= CP-Ca.H_P \qquad \text{(Eq.2.30)}$$

$$\text{where,} \; CP= Q_A+ \frac{g.A}{a} . H_A - \frac{f. Dt}{2.D.A} Q_A |Q_A|$$

$$\text{and} \quad Ca = \frac{g.A}{a}$$

Similarly negative characteristic equation may be written along the characteristic line BP,

$$Q_P = Q_B- \frac{g.A}{a}.H_B - \frac{f. Dt}{2.D.A} Q_B|Q_B| + \frac{g.A}{a} . H_P$$

$$\text{or} \; Q_P = Cn+Ca.H_P \qquad \text{(Eqs.2.31)}$$

$$\text{where,} \; Cn = Q_B - \frac{g.A}{a} . H_B - \frac{f. Dt}{2.D.A} .Q_B|Q_B|$$

$$\text{and} \quad Ca = \frac{g.A}{a}$$

In Fig. 2.3 characteristic lines AP and BP have slopes $\pm$ 1/a. Physically, these lines represent path transversed by a disturbance. In Eqs.2.30 and 2.31, there are two unknowns, namely, H_p and Q_p. The values of these unknows can be obtained by simultaneously solving these equations, i.e.,

$$Q_p = 0.5\ (CP + Cn) \qquad \text{(Eq. 2.32)}$$

Knowing Q_p, the value of H_p can be obtained either by Eqs. 2.30 or 2.31. Thus using Eqns. 2.30 and 2.31 conditions at all interior points at the end of time step can be determined.

For the calculation at downstream boundary, H_p is known and it is equal to downstream reservoir level, HRES. Q_p can be determined using positive characteristic Eq. 2.30. The flow chart of Fig. 2b, and Fig.2c illustrates the procedure for calculation.

2.8 A Computer Program for Transient Analysis of Pumps in FORTRAN Language:

A computer program for analysis of the transients in a pipeline caused by pumps is available (2). It is suitable for analysis of case1 Tracy Pumping Plant without valve. Hence above program was implemented on DEC-1090 system of I.I.T. Kanpur and the implementation was tested and verified with data given by Chaudhry(2). The results agreed with the program output given in the reference and hence the implementation of the program is considered satisfactory.

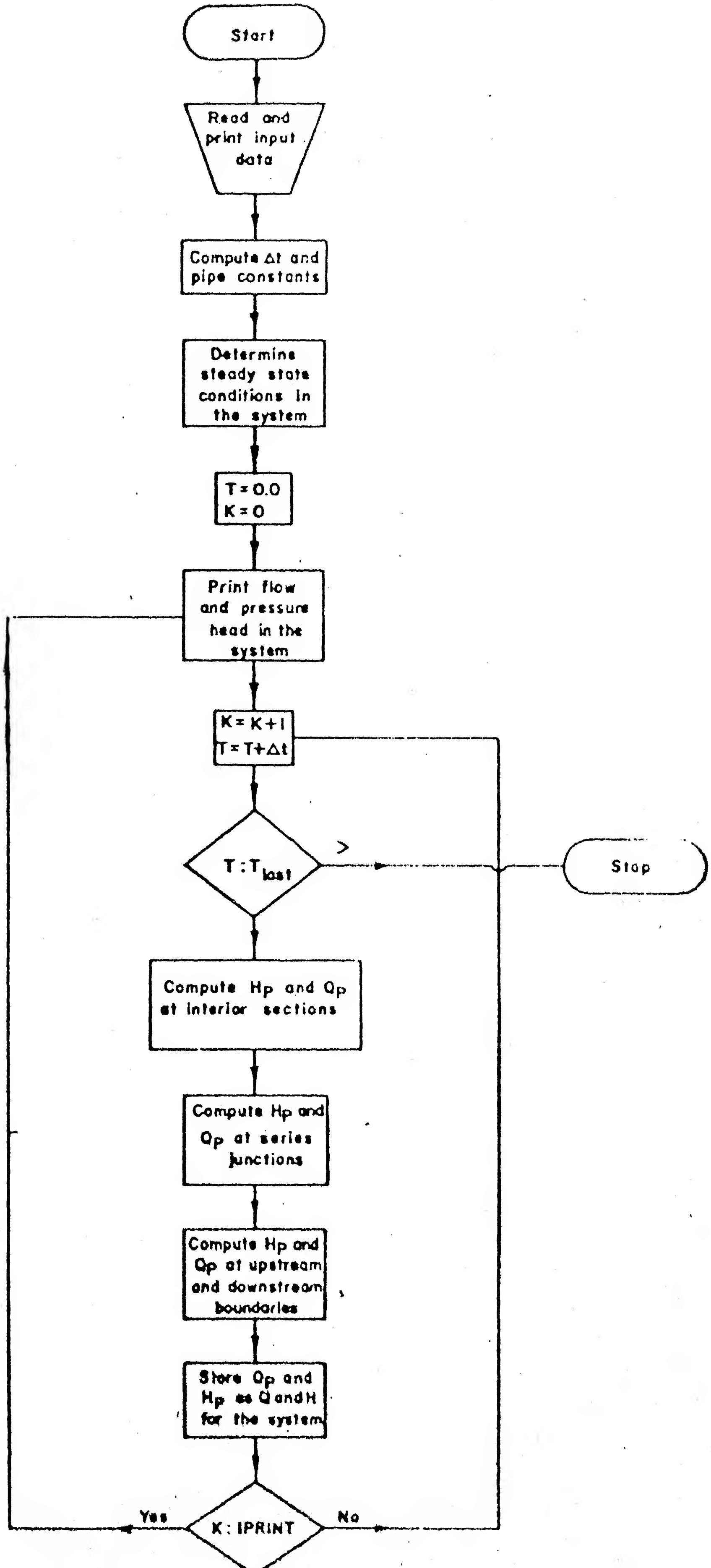

Fig. 2b Flow Chart for a Series Piping System

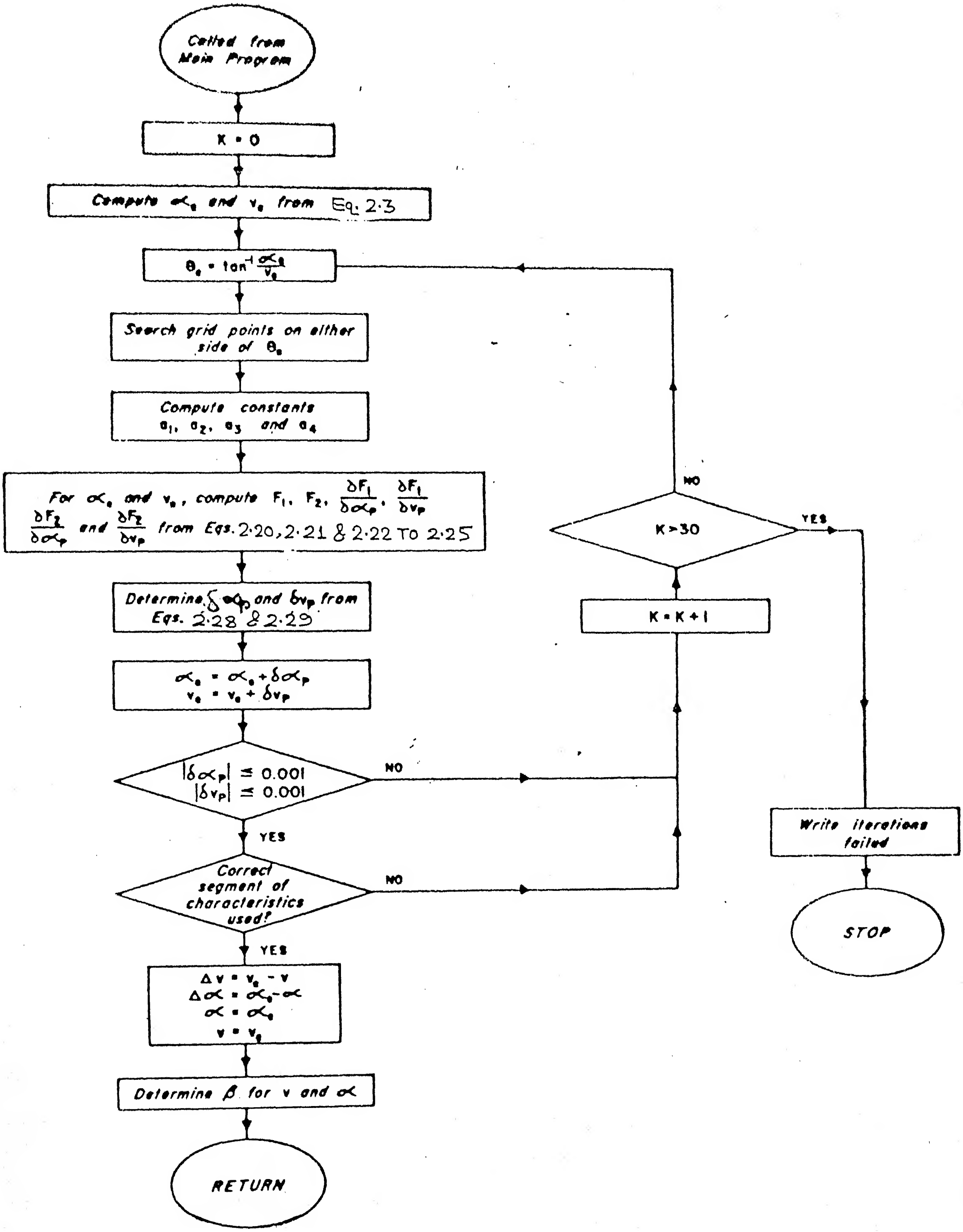

Fig. 2c Flow Chart for Boundary Conditions For Pump

2.9 Results for Case 1:

The results obtained using the data for the Tracy Pumping Plant without valve and analysed with the computer program of Chaudhry (2). The results obtained are plotted as, (h vs t), (v vs t), and (α vs t) in Fig. 2.4. Fig.2.5 represents, the transient pressure gradient in the discharge line as a result of power failure. It is found from the graph that maximum head obtained is $1.4H_R$, which is the same as obtained by Parmakian (6).

2.10 Results of the Present Study and Comparison of the Results Presented by Streeter:

Case 2, has the same plan, sectional elevation and basic data as for case 1 except for the fact that there is a butterfly valve just downstream of the pump. The method of solution is same as in case 1. except after 14 sec. Butterfly valve closure curve represented in Fig.2.6 has to be taken into account. The values of τ for different time steps has been interpolated by using parabolic interpolation.

The graphs plotted for this case are, (h vs t), (v vs t) and (α vs t) as shown in Fig. 2.6. Another graph has been plotted for this case as shown in Fig. 2.7. representing transient pressures in the discharge line as a result of power failure. It can be seen from Fig. 2.6 that maximum head is 1.3 H_R. This value indicates that there is decrease of 9% of the maximum head as obtained without butterfly valve.

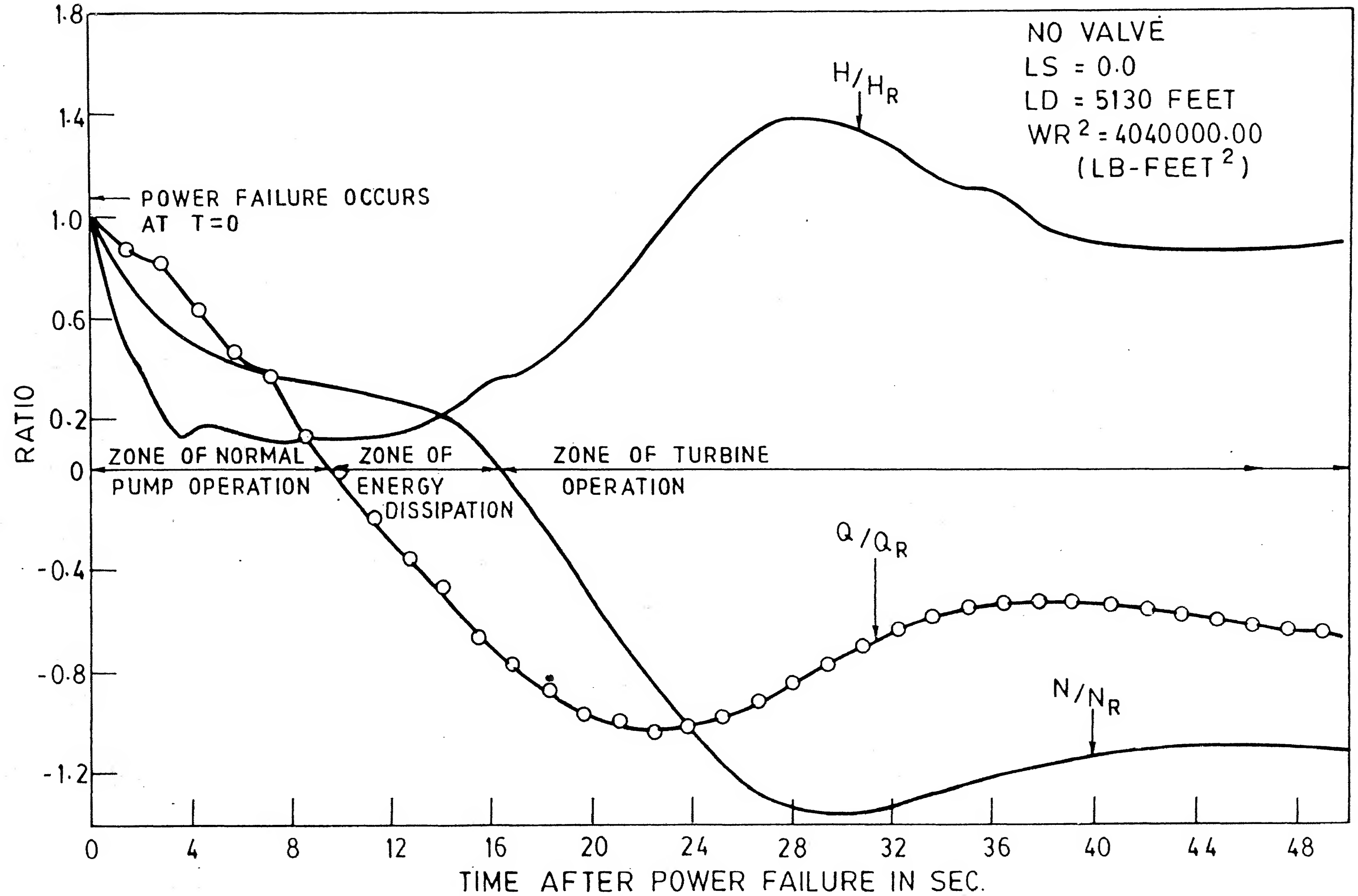

FIG. 2.4 TRANSIENT CONDITION AS A RESULT OF POWER FAILURE FOR CASE 1

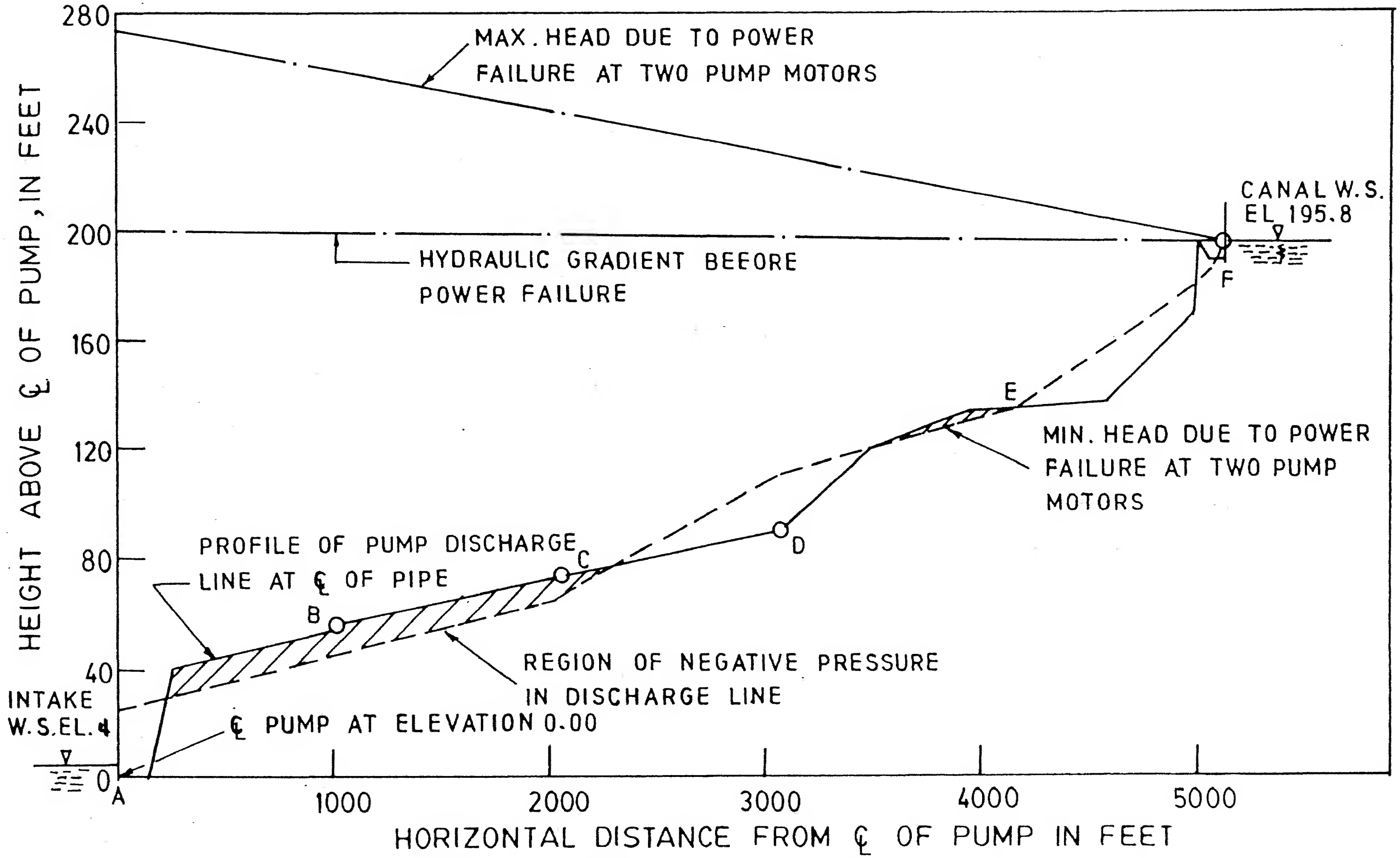

FIG. 2·5 TRANSIENT PRESSURES IN THE DISCHARGE LINE AS A RESULT OF POWER FAILURE FOR CASE 1

According to Parmakian (9) this decrease may be up to 20% if proper closing of butterfly valve is done. The pressure drop at the pump is nearly the same as that which would result without valve and will occur in 3.5 sec. after power failure of the pump motors; a complete reversal of flow in the discharge line will follow in approximately 9.8 sec. Therefore , valves with a slow time of closure can be of no value in controlling hydraulic transient conditions, because the maximum pressure rise, the maximum drop in head, and the reversal of flow in the pump discharge line will occur within a short period of time. It is found that difference between computed and observed results in normal zone of operation is negligible for which correct data has been supplied by the manufacturer. Since assumed data has been used in energy dissipation and turbine operation zone, difference is appreciable. But plotted result for (v vs t) graph obtained in this study is much closer to that obtained by Parmakian (9). Comparing Fig. 2.5 and Fig.2.7 it is observed that region of negative pressure has got reduced.

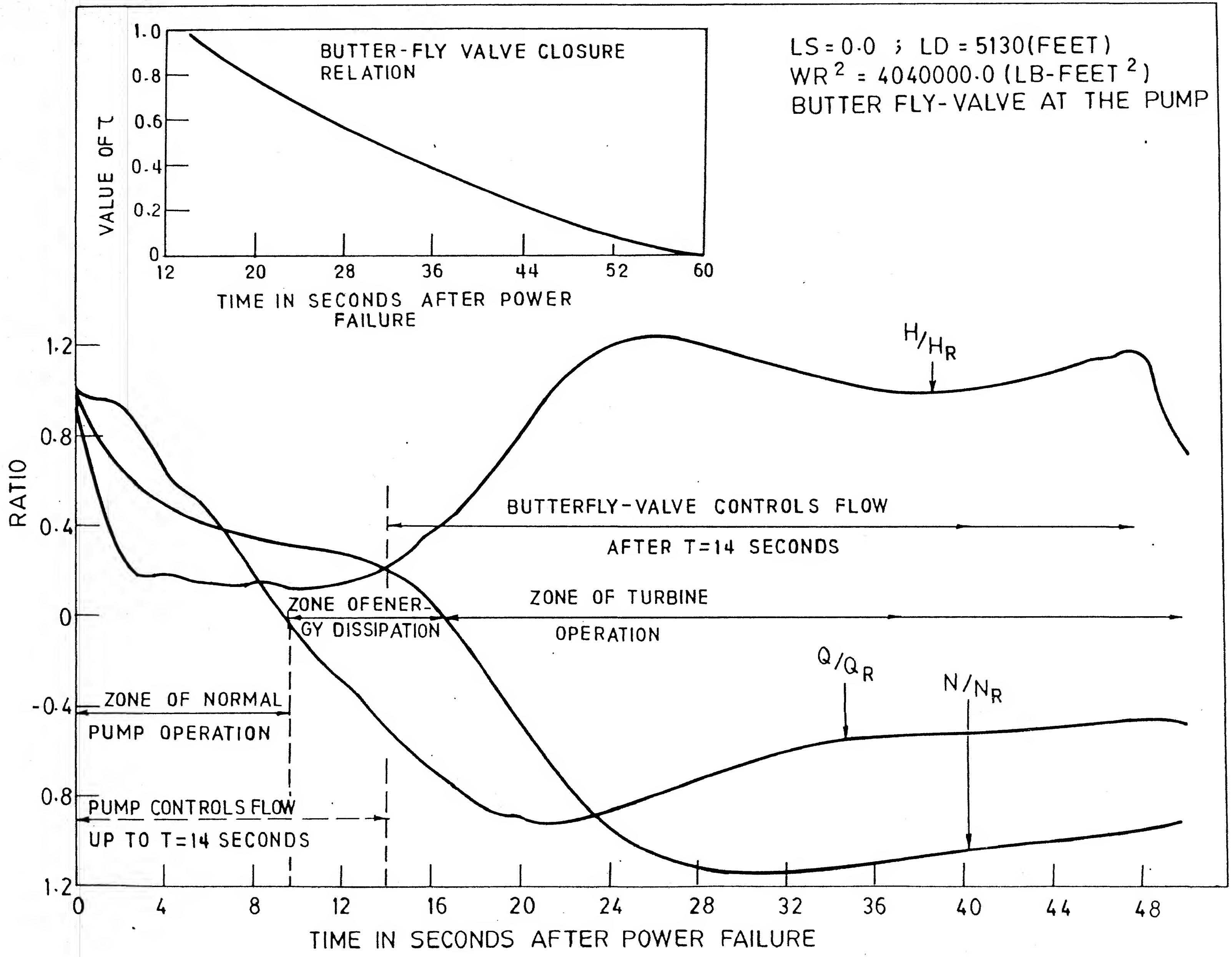

FIG. 2.6 TRANSIENT CONDITION AS A RESULT OF POWER FAILURE FOR CASE 2

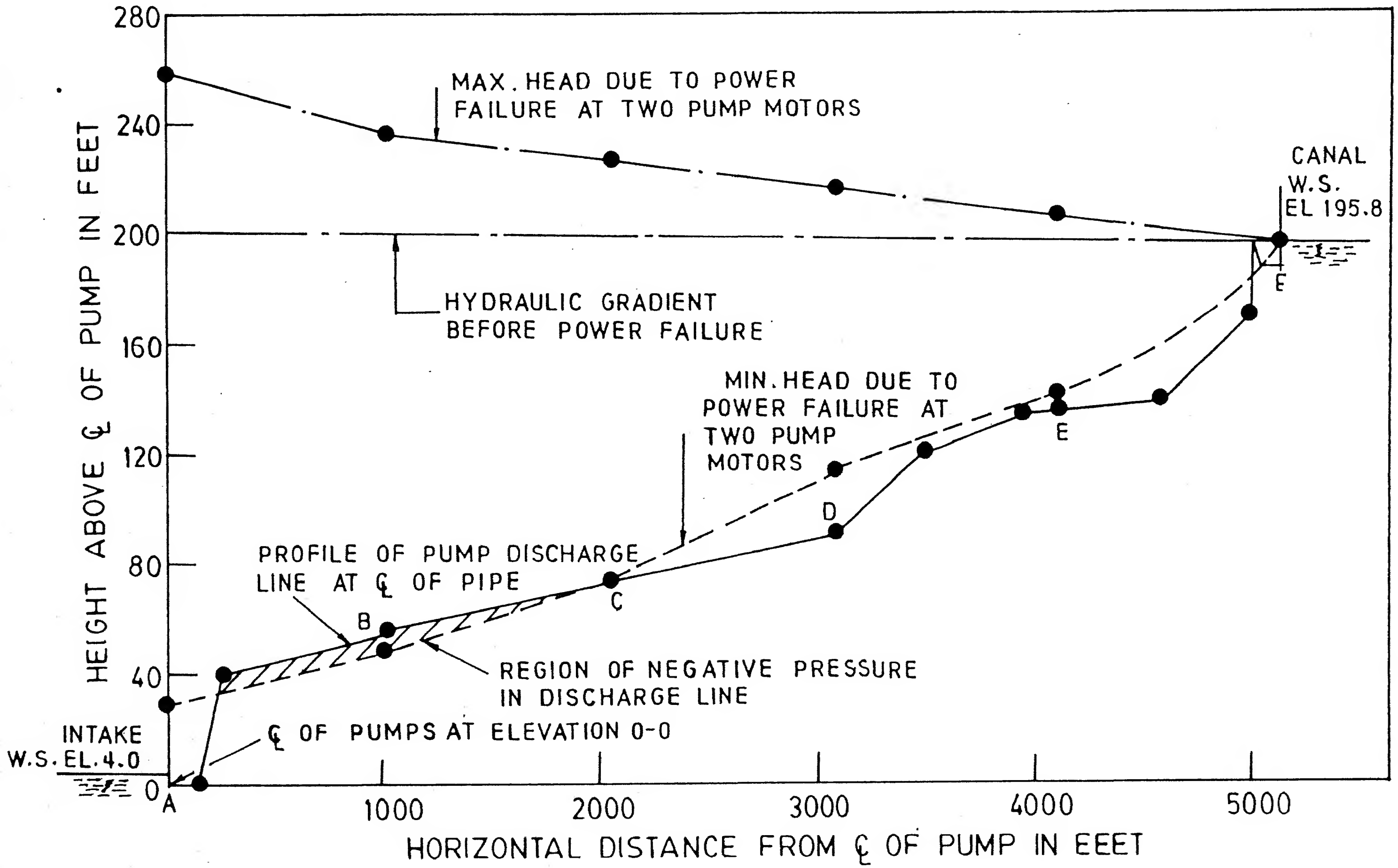

FIG. 2.7 TRANSIENT PRESSURES IN THE DISCHARGE LINE AS A RESULT OF POWER FAILURE FOR CASE 2

CHAPTER 3

ANALYSIS OF TRANSIENTS OF THE TRACY PUMPING PLANT WITH MODIFICATIONS IN THE SYSTEM

In order to study the transients at Tracy Pumping Plant with modifications in the system the following special cases were studied in detail viz., case 3 a non-return valve at the pump, case 4 a non-return valve at the pump and intermediate non-return valve at the middle of discharge line, case 5 lift dominating with two different moments of inertia of rotating parts and case 6 resistance dominating with two different moments of inertia of rotating parts. The detailed data for these cases are also shown in Table 2.2

3.1 Case 3 Tracy Pumping Plant with a non-return valve:

Case 3 has the same plan, sectional elevation and basic data as for case 1 except for the fact that there is a non-return valve just downstream of the pump. The method of solution is same as in case 1, in addition boundary condition at the pump changes when flow reverses. As flow reverses, the check valve gets closed and pump characteristics vanish from the computations.

The graphs plotted for this case are, (h vs t), (v vs t), and (α vs t) as shown in Fig. 3.1, which represent transient as a result of power failure. Another graph has been plotted for this case shown in Fig. 3.2, which shows the

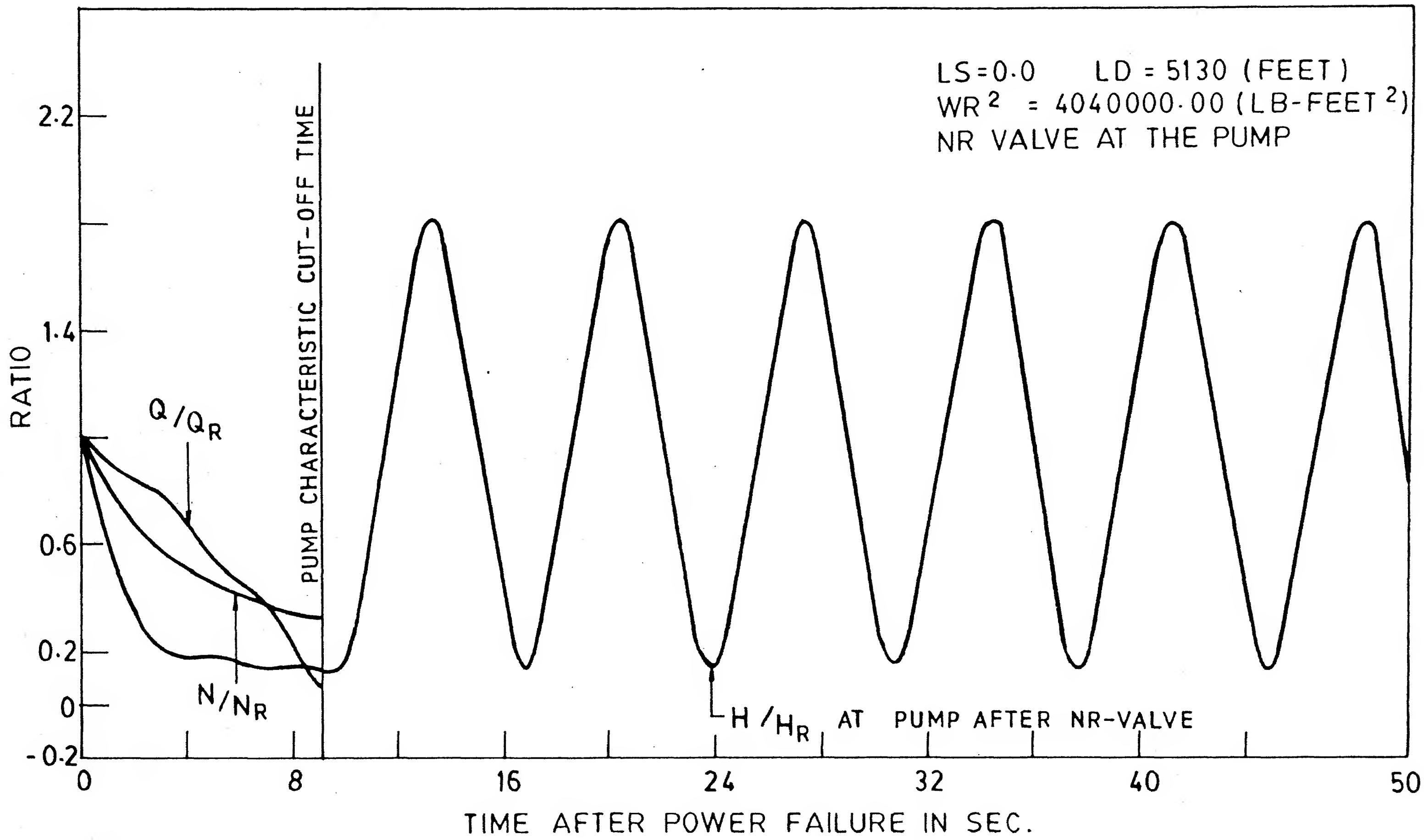

FIG. 3.1 TRANSIENT CONDITION AS A RESULT OF POWER FAILURE FOR CASE 3

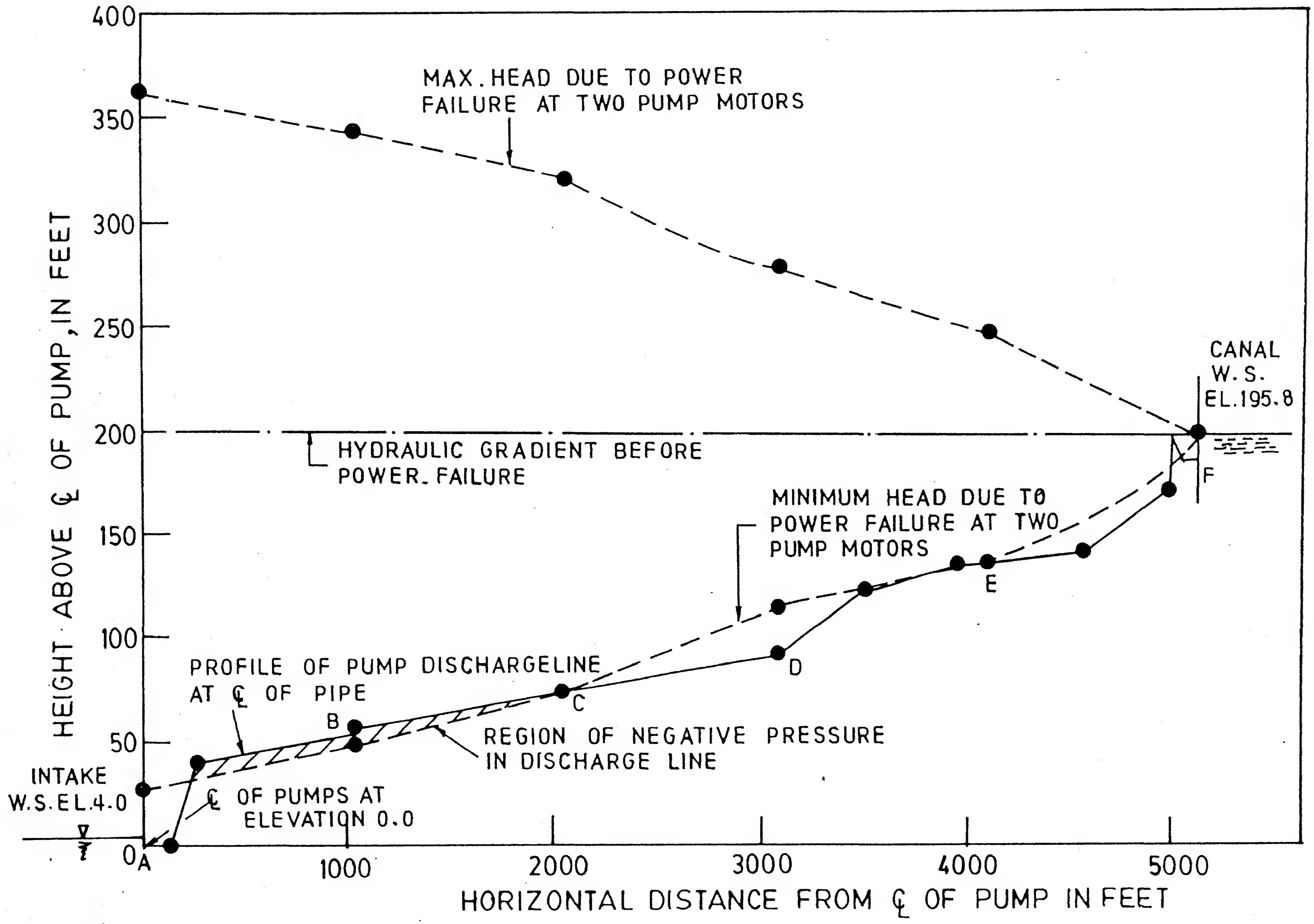

FIG. 3.2 TRANSIENT PRESSURE IN THE DISCHARGE LINE AS A RESULT OF POWER FAILURE FOR CASE 3

transient pressure in the discharge line as a result of power failure. It is found from Fig. 3.1 that head rise at the pump has gone as high as 1.9 H_R, and minimum pressure is $0.14H_R$. As discharge line is not too long and frictional effect is less, it is a case of gravity loading. Thus, for the gravity loading with all pumps failing, with undamped non-return valve, very severe fluctuating transients occur as seen from Fig.3.1, the water in the discharge line acts as a liquid spring with the hydraulic grade line at the valve oscillating about downstream reservoir. Since transient are so severe for complete failure of pump motors for gravity loadings, special precautions should be taken. Fig. 3.2 shows that the region of negative pressure for this case has got reduced as compared with case 1 and case 2, so chances of column separation has also got reduced.

3.2 Case 4 Tracy Pumping Plant with two non-return Valve:

The problem studied under case 4, is exactly same as in case 3, but in addition to case 3, there is an intermediate non-return valve at the mid section of the discharge line. For analysis of the problem the discharge line has been treated as two pipes in series, the length of the first pipe is considered upto mid section and second pipe starts from the mid section to the downstream reservoir. In the analysis of the problem three sections are of particular interest, first at the pump, second the section just before intermediate non-return valve. The boundary condition at the mid section is changed when

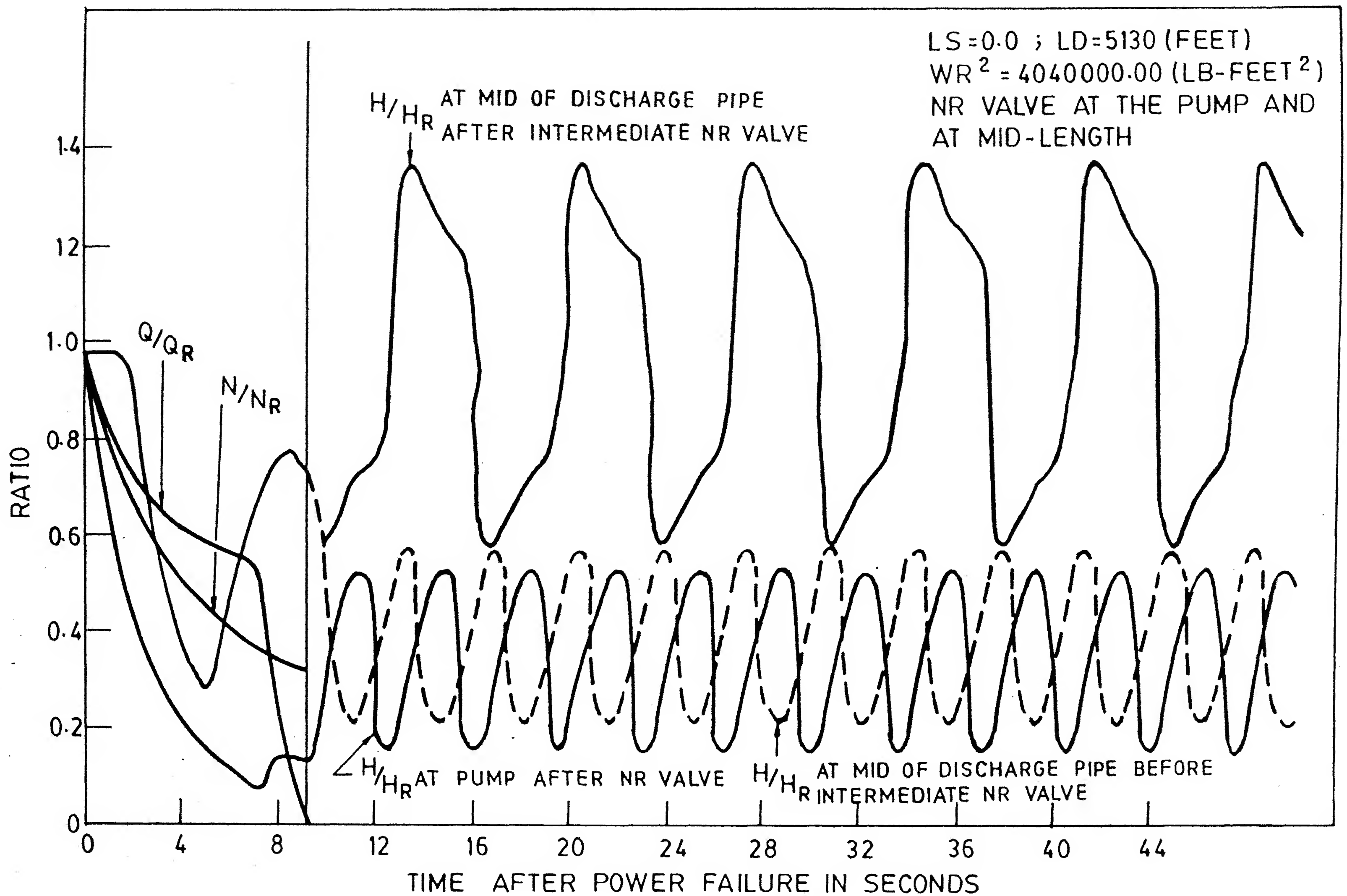

FIG. 3.3 TRANSIENT CONDITION AS A RESULT OF POWER FAILURE FOR CASE 4

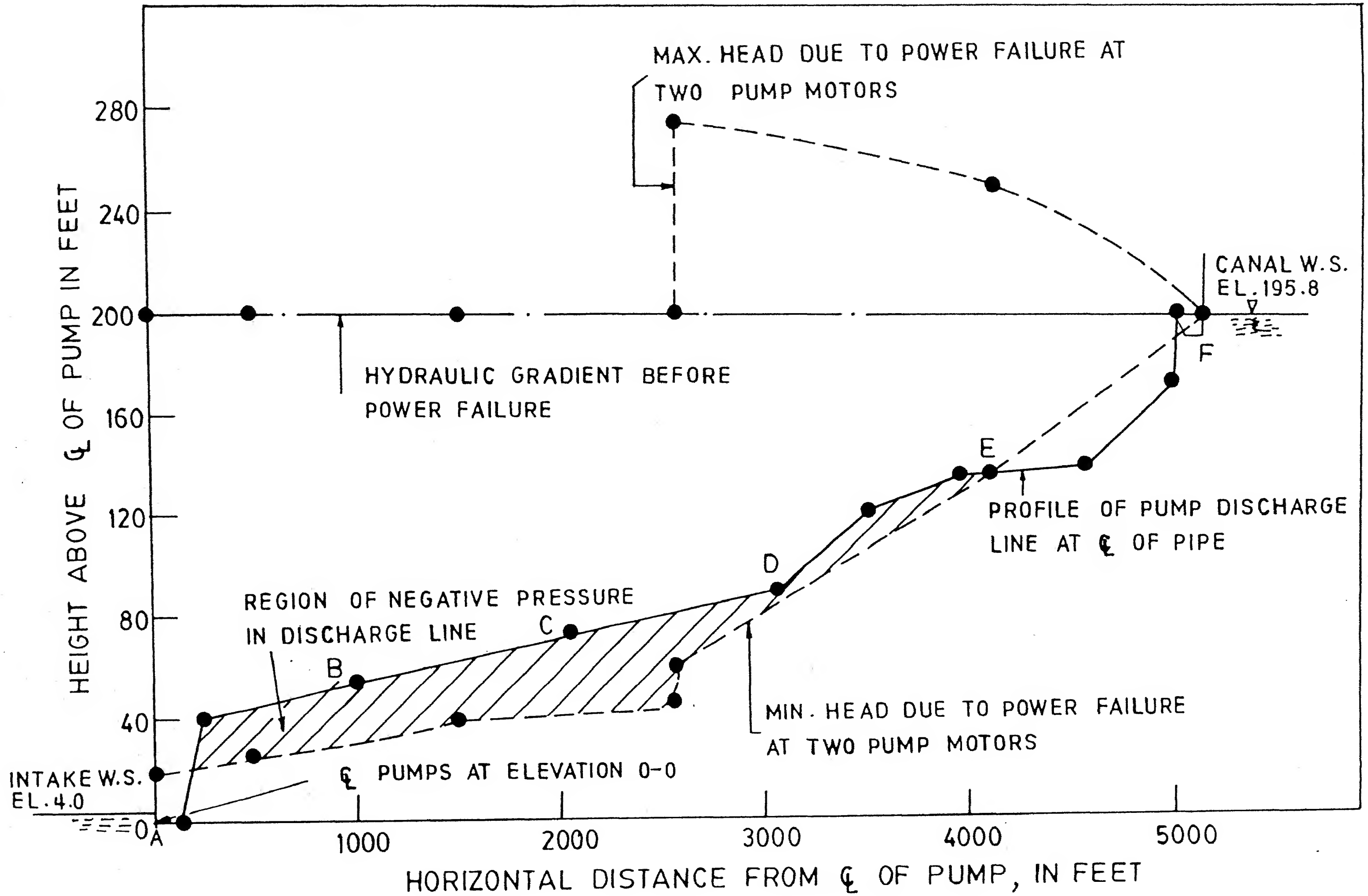

FIG. 3.4 TRANSIENT PRESSURE IN THE DISCHARGE LINE AS A RESULT OF POWER FAILURE FOR CASE 4

flow reversed, as intermediate non-return valve gets closed at that instant. The positive characteristic equation has to be used for section second and negative characteristics equation has to be used for section three. The solution procedure is same as for the case 3.

The variation of head at these sections are plotted as shown in Fig. 3.3, and it is observed that the fluctuations in head are not so severe and sharp as observed in case 3. The maximum pressure developed at the pump is nearly rated pumping head. At the second section the maximum head developed is the steady-state head, but head rise at the third section is as high as 1.39 H_R, nearly the same maximum head as in case 1. Special auxiliary equipments has to be provided whenever intermediate non-return valve has to be used. Another graph has been plotted as shown in Fig. 3.4, which shows the transient pressure in the discharge line as a result of power failure. It is observed that region of negative pressure has been increased as compared to other cases.

3.3 Pumps with Long Suction Length:

If the suction line is not short compared to discharge line, then water-hammer in the former has to be considered in the analysis. Therefore, characteristic equation for the suction line has to be included as follows

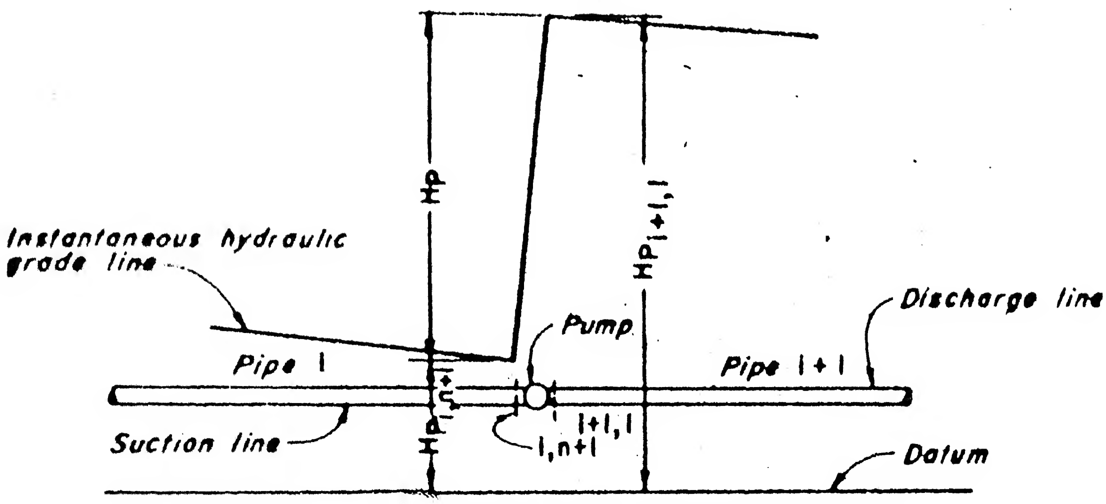

Fig 3.5 Pump with long suction line.

Refering to Fig. 3.5

$$H_P = H_{P_{i+1,1}} - H_{P_{i,n+1}} \quad \text{(Eq.3.1)}$$

$$Q_{P_{i,n+1}} = CP - Ca_i \cdot H_{P_{i,n+1}} \quad \text{(Eq.3.2)}$$

$$Q_{P_{i+1,1}} = Cn + Ca_{i+1} \cdot H_{P_{i+1,1}} \quad \text{(Eq.3.3)}$$

$$Q_{P_{i,n+1}} = Q_{P_{i+1,1}} = n_P \cdot Q_P \quad \text{(Eq.3.4)}$$

multiplying Eq.3.2 by ca_{i+1} and Eq.3.3 by Ca_i and substituting for $Q_{P_{i+1,1}}$ and $Q_{P_{i,n+1}}$ .and adding

$$n_P \cdot Q_P(Ca_i + Ca_{i+1}) = Cn.Ca_i + CP.Ca_{i+1} \cdot H_P \quad \text{(Eq.3.5)}$$

by using Q_R and H_R as reference values, Eq.3.5 may be written as

$$h_P = \frac{n_P(Ca_i + Ca_{i+1})\ Q_R \cdot v_P - Cn.Ca_i - CP.Ca_{i+1}}{Ca_i \cdot Ca_{i+1} \cdot H_R} \quad \text{(Eq.3.6)}$$

by eliminating h_p from Eqs. 4 and 3.6

$$F_1 = a_1(\alpha_P^2 + v_P^2) + a_2(\alpha_P^2 + v_P^2) \tan^{-1}\frac{\alpha_P}{v_P} - C_7 \cdot v_P + C_8 = 0 \quad \text{(Eq.3.7)}$$

in which $$C_7 = \frac{n_P \cdot Q_R(Ca_i + Ca_{i+1})}{Ca_i \cdot Ca_{i+1} \cdot H_R} \quad \text{(Eq.3.8)}$$

and $$C_8 = \frac{Cn.Ca_i + CP.Ca_{i+1}}{Ca_i \cdot Ca_{i+1} \cdot H_R} \quad \text{(Eq.3.9)}$$

$$\frac{\partial F_1}{\partial \alpha_P} = 2.a_1 \cdot \alpha_P + 2.a_2 \cdot \alpha_P \cdot \tan^{-1}\frac{\alpha_P}{v_P} + a_2 \cdot v_P \quad \text{(Eq.3.10)}$$

$$\frac{\partial F_1}{\partial v_P} = 2.a_1 \cdot v_P + 2.a_2 \cdot v_P \tan^{-1}\frac{\alpha_P}{v_P} - a_2 \cdot \alpha_P - C_7 \quad \text{(Eq.3.11)}$$

and for this case F_2, $\frac{\partial F_2}{\partial \alpha_p}$ and $\frac{\partial F_2}{\partial v_p}$ are given by Eqs. 2.21, 2.24 and 2.25.

In the study of transients in pipelines due to failure of pumps two different cases can be identified, one in which the resistance of the pipeline is small and hence the lift is dominating, and the other where the lift is small and resistance is dominating since the transients in the pipelines under these two cases are very different it is proposed to study them separately.

3.4 Case 5 Pipes with Lift Dominating:

The system studied has 1282.5 ft. of suction pipe and 3847.5 ft. of discharge pipe with a head loss due to resistance under steady-state of 3.9 ft. which is small in comparison to the gravity lift of 197.1 ft.

3.4.1 Case 5 a. Small Moment of Inertia of Rotating Parts:

This case takes into account the length of suction pipe, and as a result characteristic equation for the suction line has to be taken into account.

As suction length has not been neglected, the expression for F_1, $\frac{\partial F_1}{\partial \alpha_p}$, and $\frac{\partial F_1}{\partial v_p}$ taken for case 1 is now changed. The expressions for these variables F_1, $\frac{\partial F_1}{\partial \alpha_p}$, and $\frac{\partial F_1}{\partial v_p}$ are defined by the Eqs. 3.7, 3.10 and 3.11 respectively. These expressions has two constants C_7 and C_8 which have been

defined by the Eqs. 3.8 and 3.9. The solution procedure adopted is same as for case 1.

The graphs plotted for this case are, (h vs t), (v vs t), and (α vs t) as shown in Fig. 3.6, which represent transient condition as a result of power failure. The maximum steady-state head is 201 ft. which goes as high as 275.2 ft.

3.4.2 Case 5b Large Moment of Inertia of Rotating Parts:

This modification in the Tracy Pumping Plant has been studied under case 5b. of the computer program. This case is exactly same as case 5a, except moment of inertia of rotating parts has been doubled, which is being equal to 8080000 pound feet squared. The equations involved are same as for case 5a and solution procedure adopted is same as for case 5a.

The graphs plotted for this case are, (h vs t), (v vs t), and (α vs t) as shown in Fig. 3.7, which represent transient conditions as a result of power failure. The maximum steady-state head is 201 ft. which goes as high as 251.25 ft. The effect of moment of inertia can be seen from Fig. 3.8. It is found that doubling the moment of inertia lowers the maximum head developed along the discharge line but increases minimum head along the discharge line. The friction drop is very small for case 5a and hence loading is due to gravity lift. It is observed that increase

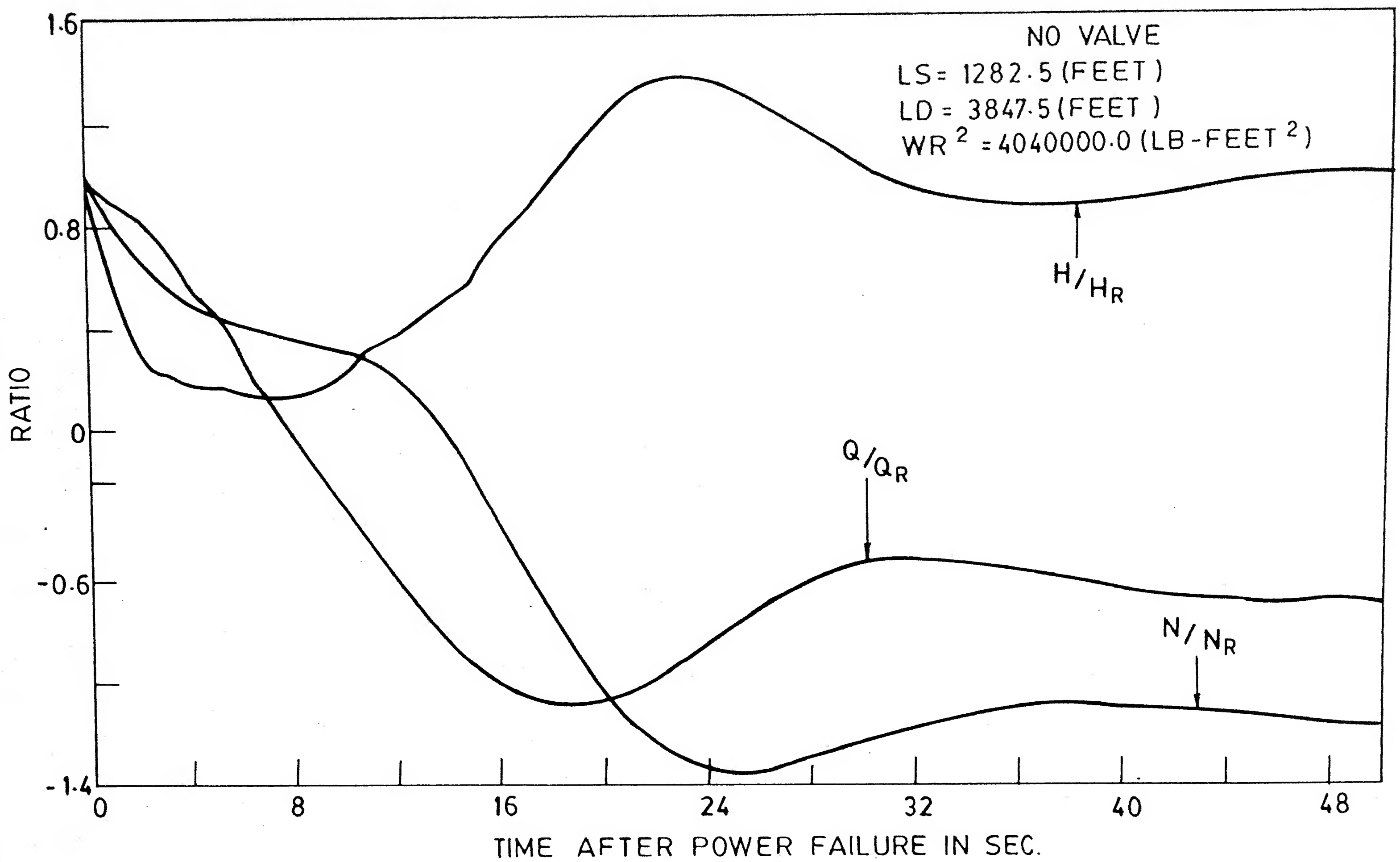

FIG. 3.6 TRANSIENT CONDITION AS A RESULT OF POWER FAILURE FOR CASE 5a

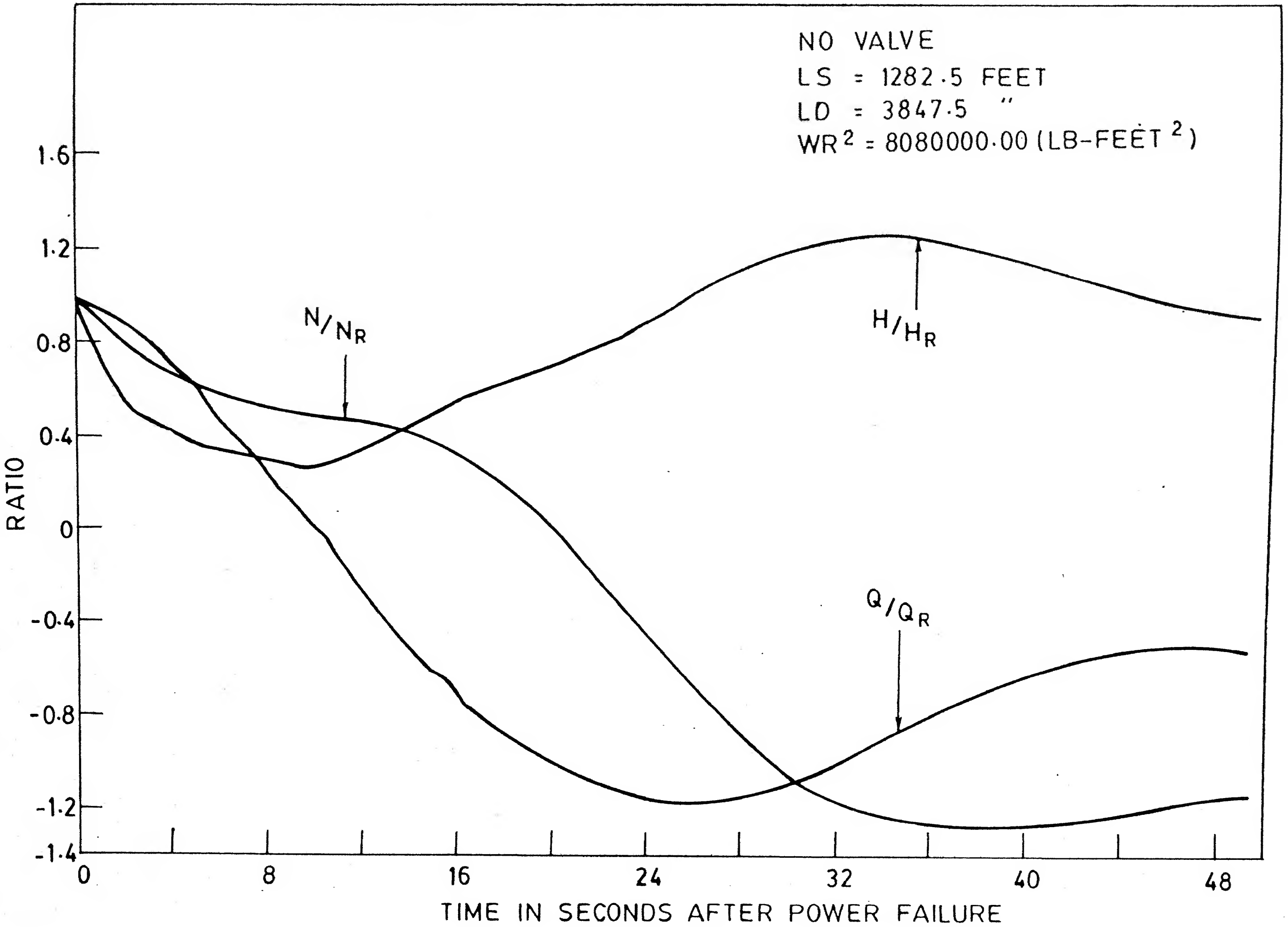

FIG. 3.7 TRANSIENT CONDITION AS A RESULT OF POWER FAILURE FOR CASE 5b

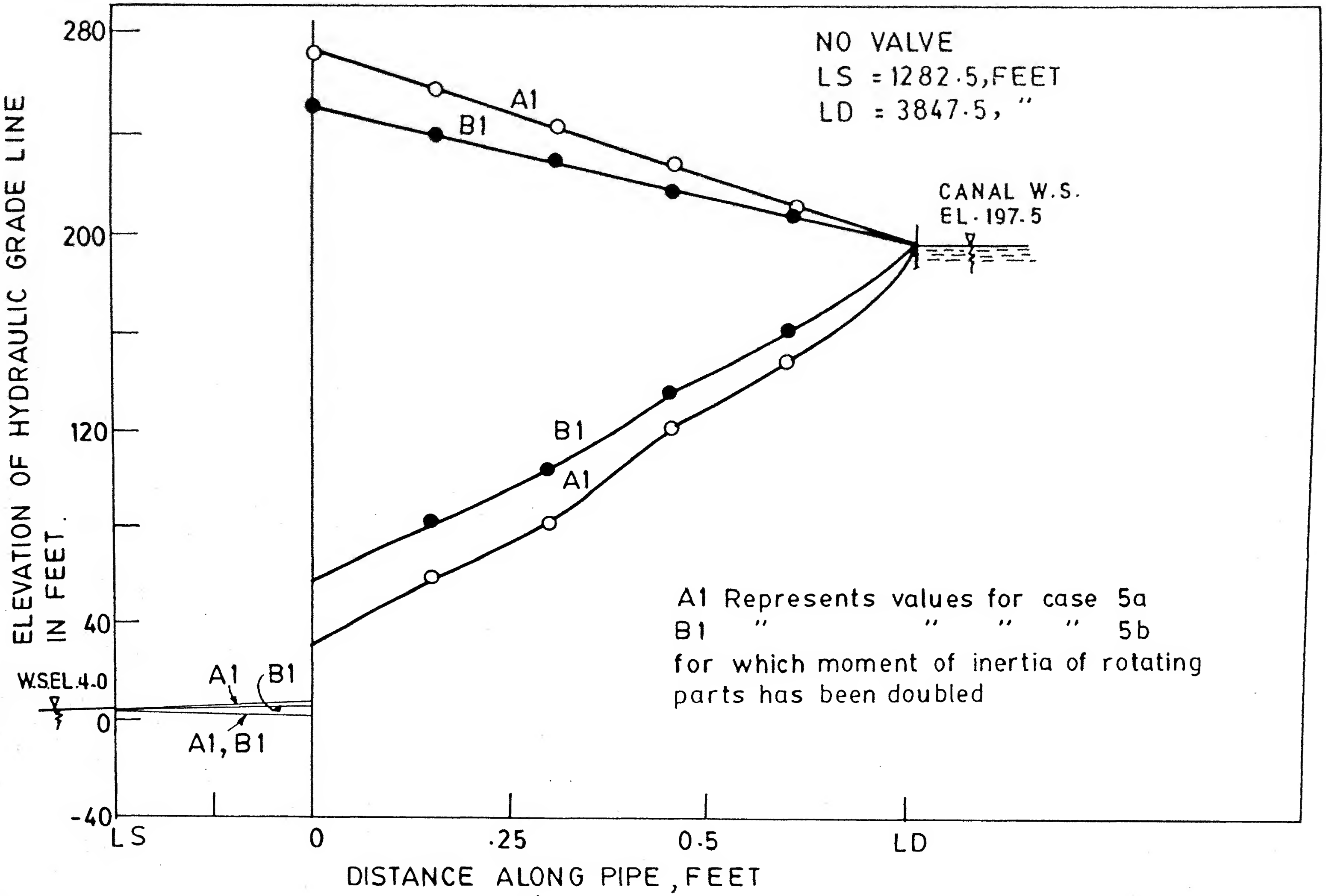

FIG. 3.8 MAX. AND MIN. VALUES OF HYDRAULIC GRADE LINE FOR SIMULTANEOUS FAILURE OF TWO PUMPS WITHOUT DISCHARGE VALVE IN AN 15 FT. DIA. LINE. COMPARISON OF CASE 5a AND 5b

in moment of inertia of rotating parts will be benificial.

3.5 Case 6 Pipe Line with Resistance Dominating:

The system studied has suction pipe 3847.5 ft. long and discharge pipe 23085 ft. long with a head loss due to resistance under steady-state of 23.4 ft. and a gravity lift of 177.6 ft. As discharge line length is very large frictional loss is considerable, and naturally this case may be called as resistance dominating. The equations involved and method of solution is exactly same as in case 5a or case 5b.

3.5.1 Case 6a Small Moment of Inertia of Rotating Parts:

The graphs plotted for this case are, (h vs t), (v vs t), and (α vs t) as shown in Fig. 3.9,which represent transient conditions as a result of power failure. It can be observed that the maximum transient head equals the steady-state head.

3.5.2 Case 6b Large Moment of Inertia of Rotating Parts:

In this case all the data taken are same as for case 6a. except the moment of inertia which has been doubled to 8080000pound-ft. squared. The method of solution is same as for case 6a and equations involved are also the same.

The graphs plotted for this case are, (h vs t), (v vs t), and (α vs t), which represent transient conditions as result of power failure as shown in Fig. 3.10. It can be

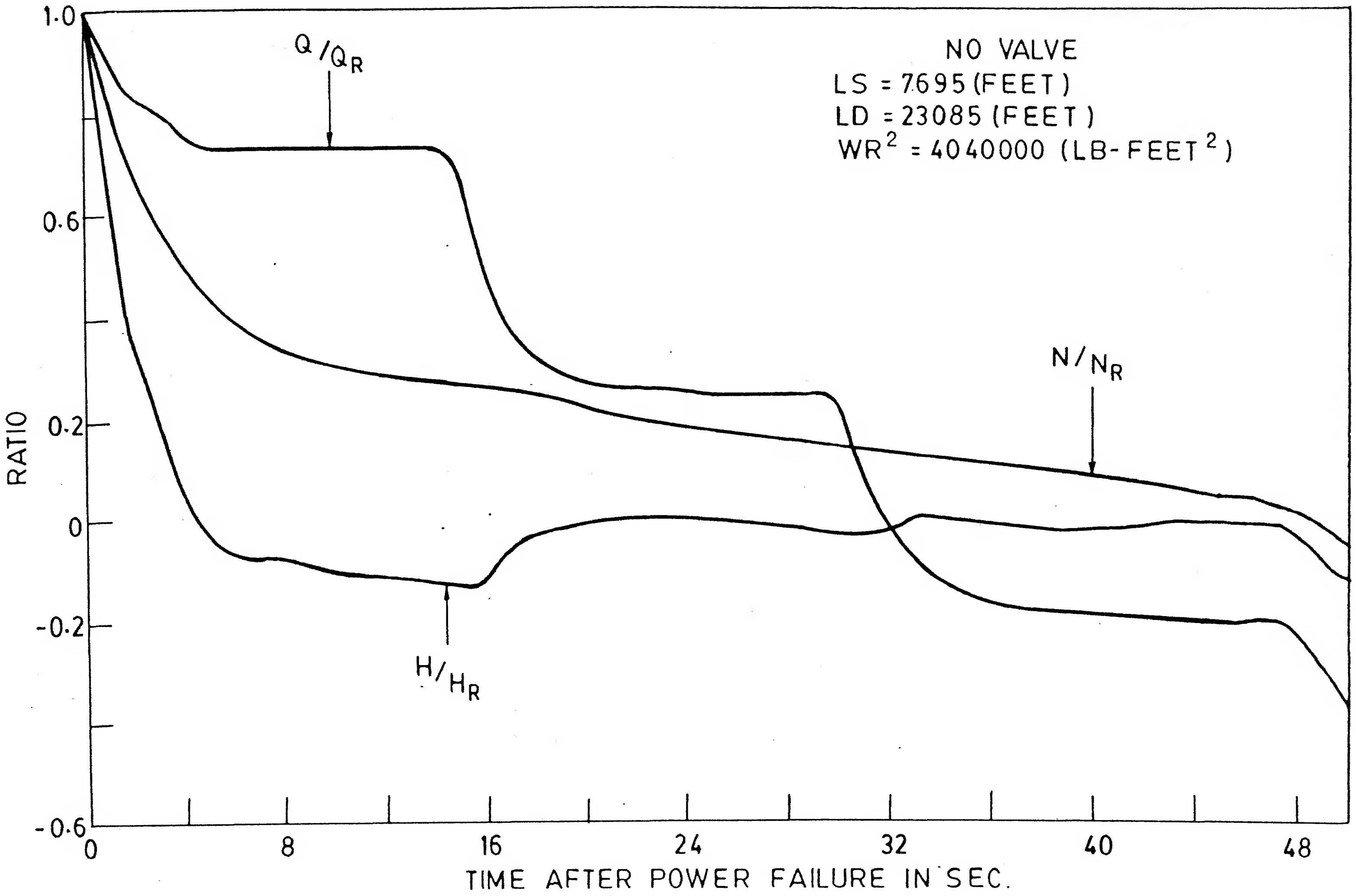

FIG. 3.9 TRANSIENT CONDITION AS A RESULT OF POWER FAILURE FOR CASE 6a

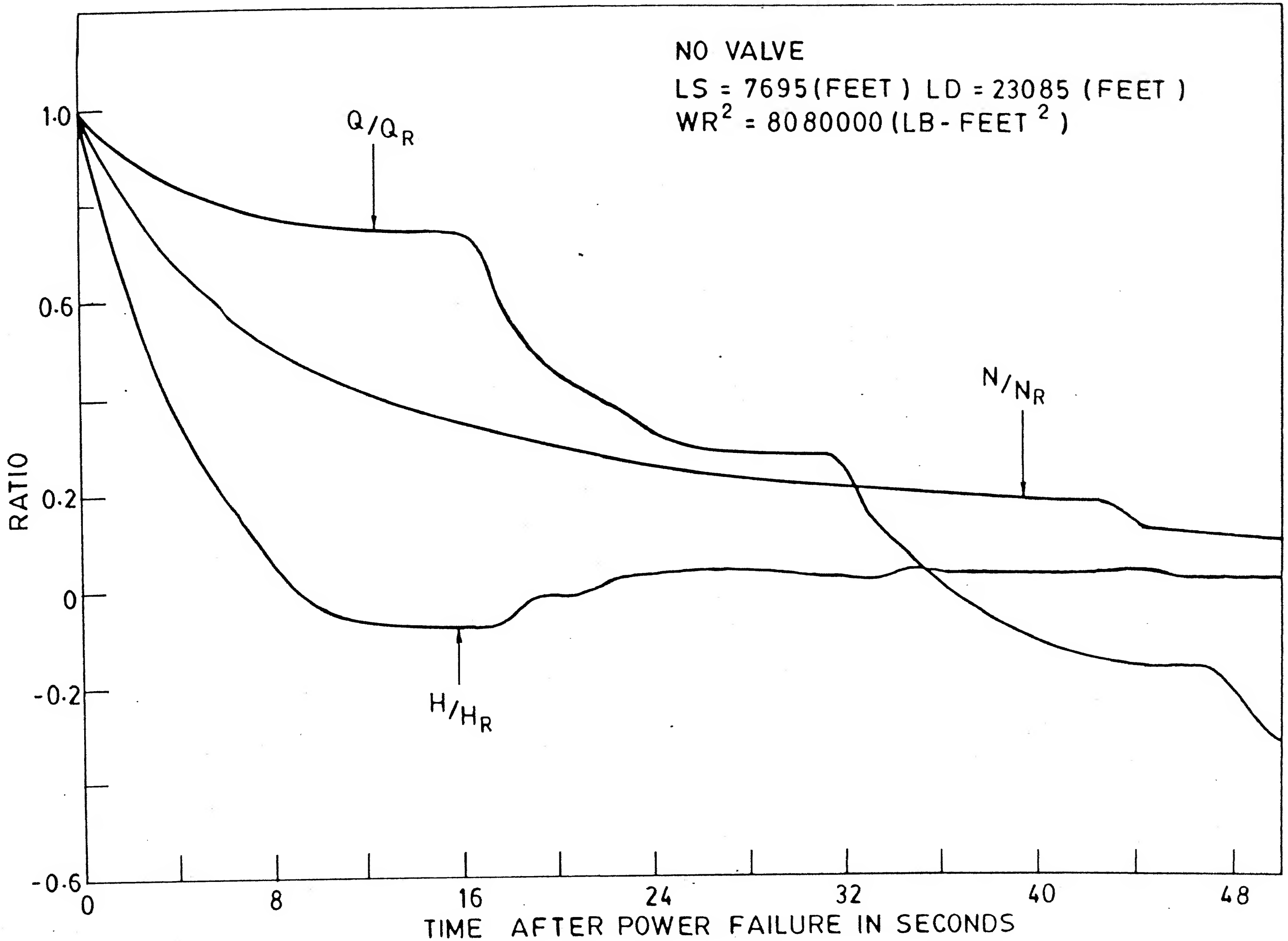

FIG. 3.10 TRANSIENT CONDITION AS A RESULT OF POWER FAILURE FOR CASE 6b

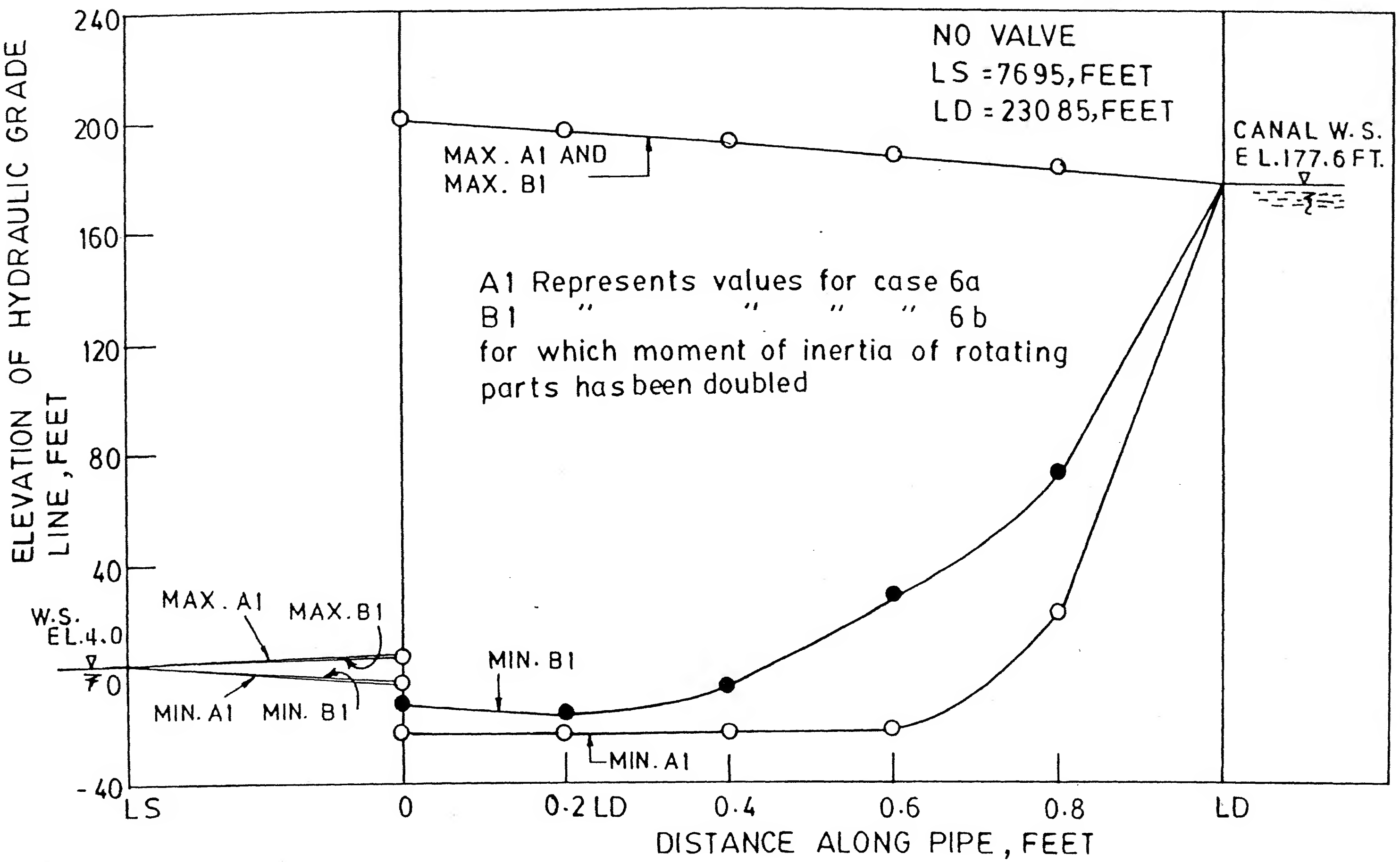

FIG. 3.11 MAX. AND MIN. VALUES OF HYDRAULIC GRADE LINE FOR SIMULTANEOUS FAILURE OF TWO PUMPS WITHOUT DISCHARGE VALVE IN AN 15 FEET DIA. LINE COMPARISON OF CASE 6a AND 6b

CHAPTER 4

ANALYSIS OF THE TRANSIENTS OF THE TRACY PUMPING PLANT WITH AIR CHAMBER

4.1 Introduction:

An air chamber is a vessel having compressed air at its top and connected to the discharge pipe.Fig.4.1. An orifice is provided to restrict the inflow into or outflow from the chamber. Here a differential orifice has been used such that it produces more head loss for inflow into the chamber than for a corresponding outflow from the chamber. The outflow from the chamber has been made free so that column separation in the pipeline is avoided and inflow has been restricted to reduce the size of the chamber.

Analysis of the system with air chamber has been carried out because air chamber has its own advantages over other auxiliary equipments such as surge tanks. They include the following:

1) The volume of an air chamber required to keep the maximum and minimum pressures within the prescribed limits is smaller than that of an equivalent surge tank.
2) An air chamber can be installed with its axis parallel to the ground slope. This reduces the foundation costs

and provides better resistance to both wind and earthquake loads.

3) An air chamber can be provided near the pump, which may not be practical in the case of surge tank because of excessive height. This reduces the pressure rise and the pressure drop in the pipe line.

4) To prevent freezing in cold climates, it is cheaper to heat the liquid in an air chamber than in a surge tank because of smaller size and because of proximity of the pumphouse.

The main disadvantage of air chamber is the necessity to provide air compressors and auxiliary equipment, which require constant maintenance.

A ratio of 2.5:1 between the orifice head losses for the same inflow and outflow has been used. Six cases have been studied varying different parameters, $\frac{2.CO.a}{QO.L}$, in which CO is initial volume of air inside the chamber, diameter of the orifice, water level in the chamber. In the analysis of these cases with different parameters, location of air chamber has been taken as downstream of the pump, after a check valve.

4.2 Boundary Conditions for Air Chamber:

Referring to Fig.4.2, at the junction of the chamber with the pipeline following equations are availabe:

1) Positive characteristic equation for section(i,n+1):

$$Q_{P_{i,n+1}} = Cp - Ca_i \cdot H_{P_{i,n+1}} \quad \text{(Eq.4.1)}$$

2) Negative characteristic equation for section (i+1,1):

$$Q_{P_{i+1,1}} = Cn + Ca_{i+1} \cdot H_{P_{i+1,1}} \quad \text{(Eq.4.2)}$$

3) If the loses at the junction are neglected, then

$$H_{P_{i,n+1}} = H_{P_{i+1,1}} \quad \text{(Eq.4.3)}$$

4) Continuity equation:

$$Q_{P_{i,n+1}} = Q_{P_{i+1,1}} + Q_{Porf} \quad \text{(Eq.4.4)}$$

in which Q_{Porf} = flow through the orifice and is considered to be positive when flow is into the chamber .

Assuming that the air enclosed at the top of air chamber follows polystropic relation for a perfect gas, i.e.

$$H^{*}_{Pair} \cdot v^{m}_{Pair} = C \quad \text{(Eq.4.5)}$$

in which H^{*}_{Pair} and v_{Pair} are the absolute pressure head and the volume of the enclosed air at the end of time step, m is the exponent in the polystropic gas equation. The value of constant C is determined from initial stready-state conditions, i.e., $C = CO^{m} \cdot H^{*}_{Oair}$. The values of m are equal to 1.0 and 1.4 respectively for an isothermal and for an adiabatic expansion or contraction of air. For design calculations, an average value of m = 1.2 may be used because transient are usually rapid at the beginning and they are slow at the end.

The orifice losses may be expressed as

$$h_{Porf} = C_{orf} Q_{Porf} |Q_{Porf}| \qquad \text{(Eq.4.6)}$$

in which C_{orf} = coefficient of orifice loss

For the enclosed air volume following equations may be written:

$$H^{*}_{Pair} = H_{P_{i,n+1}} + H_b - Z_P - h_{Porf} \qquad \text{(Eq.4.7)}$$

$$V_{Pair} = CO - AC(Z_P - Z) \qquad \text{(Eq.4.8)}$$

$$Z_P = Z + 0.5(Q_{orf} + Q_{Porf}) \frac{Dt}{AC} \qquad \text{(Eq.4.9)}$$

in which H_b = barometric pressure head,

AC = horizontal cross-sectional area of the chamber,

Z and Z_P= heights of the liquid surface in the chamber above the datum at the beginning and at the end of the time step,

and CO = volume of air at beginning of time step.

From Eqs. 4.1,4.2 and 4.4

$$Q_{Porf} = (Cp - Cn) - (Ca_i + Ca_{i+1}) H_{P_{i,n+1}} \qquad \text{(Eq.4.10)}$$

and from Eqs. 4.5,4.6, 4.7 and 4.8.

$$(H_{P_{i,n+1}} + H_b - Z_P - C_{orf} \cdot Q_{Porf} |Q_{Porf}|) \; [\,CO - AC(Z_P - Z)\,]^m = C$$

(Eq.4.11)

Eqs. 4.9, 4.10 and 4.11 have three unknowns, namely Q_{Porf}, $H_{P_{i,n+1}}$ and Z_P. Substituting two of these equations one equation in one unknown can be derived. For example the equation written in terms of, Q_{Porf} and other known variables is as follows.

$$\therefore \text{FUN} = \left[\frac{Cp-Cn-Q_{Porf}}{Ca_i+Ca_{i+1}} + H_b - Z - \frac{0.5xDT}{AC} (Q_{orf}+Q_{Porf}) - C_{orf} \times Q_{Porf} \times |Q_{Porf}|) \right] \times$$

$$[CO-0.5 \times (Q_{orf}+Q_{Porf}) \times DT]^m - C = 0 \qquad \text{(Eq.4.12)}$$

The corresponding derivative of the function, FUN with respect to Q_{Porf} is given below:

$$\therefore \text{DFUN} = -\left[\frac{1}{Ca_i+Ca_{i+1}} + \frac{0.5xDT}{AC} + 2x\, C_{orf} \times |Q_{Porf}| \right] \times$$

$$[CO-0.5xDT\ (Q_{orf}+Q_{Porf}]^m$$

$$-\ 0.5xmxDT \left[\frac{Cp-Cn-Q_{Porf}}{Ca_i+ Ca_{i+1}} + H_b + Z - \frac{0.5xDT}{AC} (Q_{orf}+Q_{Porf}) - C_{orf} \times Q_{Porf} \times |Q_{Porf}| \right] \times$$

$$[CO-0.5xDT\ (Q_{orf}+Q_{Porf})]^{m-1} = 0 \qquad \text{(Eq.4.13)}$$

Then Newton-Raphson method can be used to evaluate, the value of the unknown Q_{Porf}. The flow chart for this procedure has been shown in Fig. 4.5.

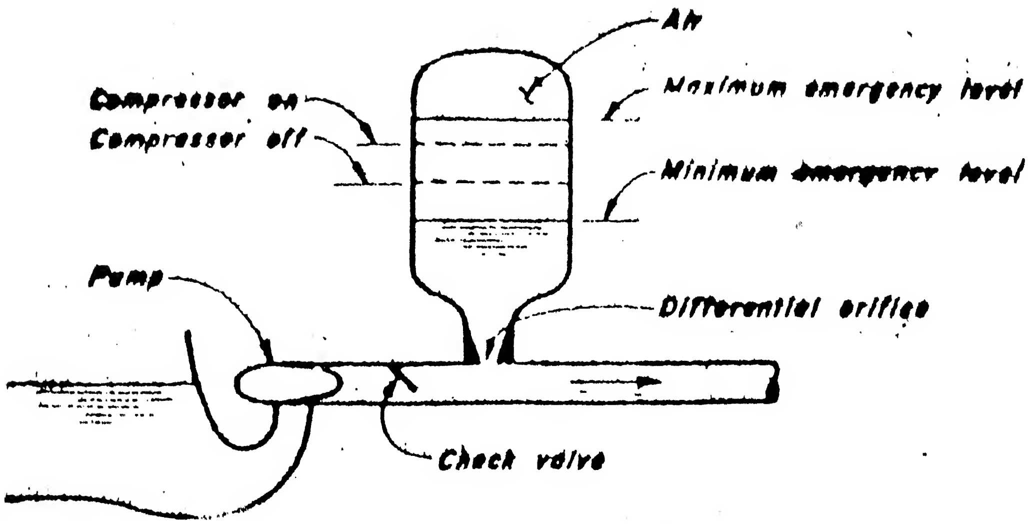

Fig 4·1 Air chamber.

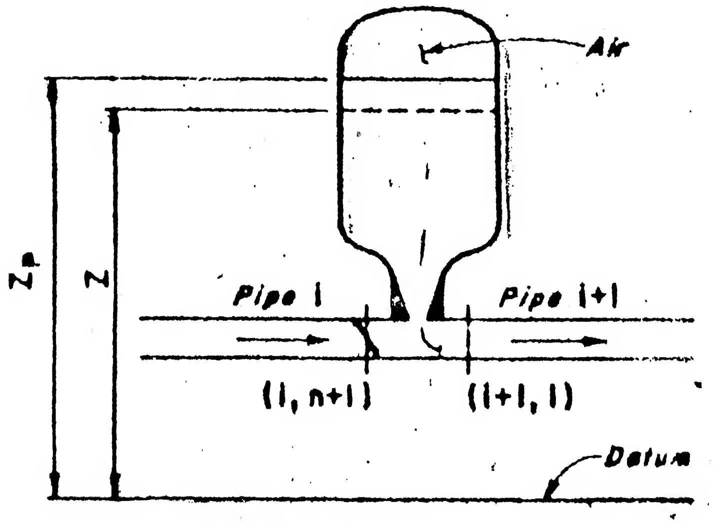

Fig 4·2 Notation for air chamber.

4.3 Parametric Analysis of Air Chamber:

Transient problem studies under case 7. has an air chamber just downstream of pump after check valve. The length of discharge line, moment of inertia of rotating parts and other datas are same as in case 1.

It has been assumed that check valve closes simultaneously with power failure, so pump characteristics vanish from calculations. The boundary conditions, at the pump with air chamber has been developed in section 4.2. The equations are simplified to have one unknown Q_{Porf} as defined by Eq. 4.12 and derivative of this equation has been defined by the Eq. 4.13. Newton-Raphson method has been used to find the root of the equation ,Q_{Porf}. By back substituting Q_{Porf} in Eq. 4.11 we get the value of $H_{P_{i,n+1}}$. Z_P can be obtained by using evaluated value of Q_{Porf} in Eq. 4.10. For the calculation at next time step Q_{orf} is assigned the value of Q_{Porf}, Z is assigned the value of Z_P, and CO is assigned the value of v_{Pair}. This procedure is repeated upto the required time. This case has been studied under six subdivisions by changing the values of different parameters involved in the computation. The different parameters involved are, the constant $\frac{2.CO.a}{QO.L}$, in which CO is initial volume of air in the chamber, diameter of the orifice and initial water level inside the chamber. The input data for different cases are shown in Table 4.1.

4.4 Discussion of Results:

Graphs have been plotted between water surface elevation inside air chamber and time as shown in Fig.4.2 and Fig. 4.4. It can be observed that in case 7c and 7e the fluctuations in water surface elevation are sharp and quick. The fluctuations about mean in case 7f are minimum and also the fluctuations are not intense and sharp.

It is observed from Table 4.1 that the effect of decreasing $\frac{2.CO.a}{QO.L}$ is that it increased upsurge and downsurge. The change in orifice diameter from 7.5 ft. to 5.0 ft. did not result in any change in the upsurge and the downsurge. The effect of decreasing the initial water surface elevation is that, maximum water surface elevation decreases and minimum water surface elevation also decreases, The minimum water surface elevations obtained in case 7c and case 7e are negative so chamber designed using parameters of these cases are unacceptable. In case 7b minimum cover of water column is less than half the diameter of discharge line so parameters used in this case are not acceptable. The total height of chamber in case 7f. comes to be lowest 83.4 ft, but inside air pressure comes to be higher than that in case 7d. in which case height of chamber comes to be 98.4 ft. The total height of the chamber, maximum upsurge and minimum downsurge obtained has been given in Table 4.1.

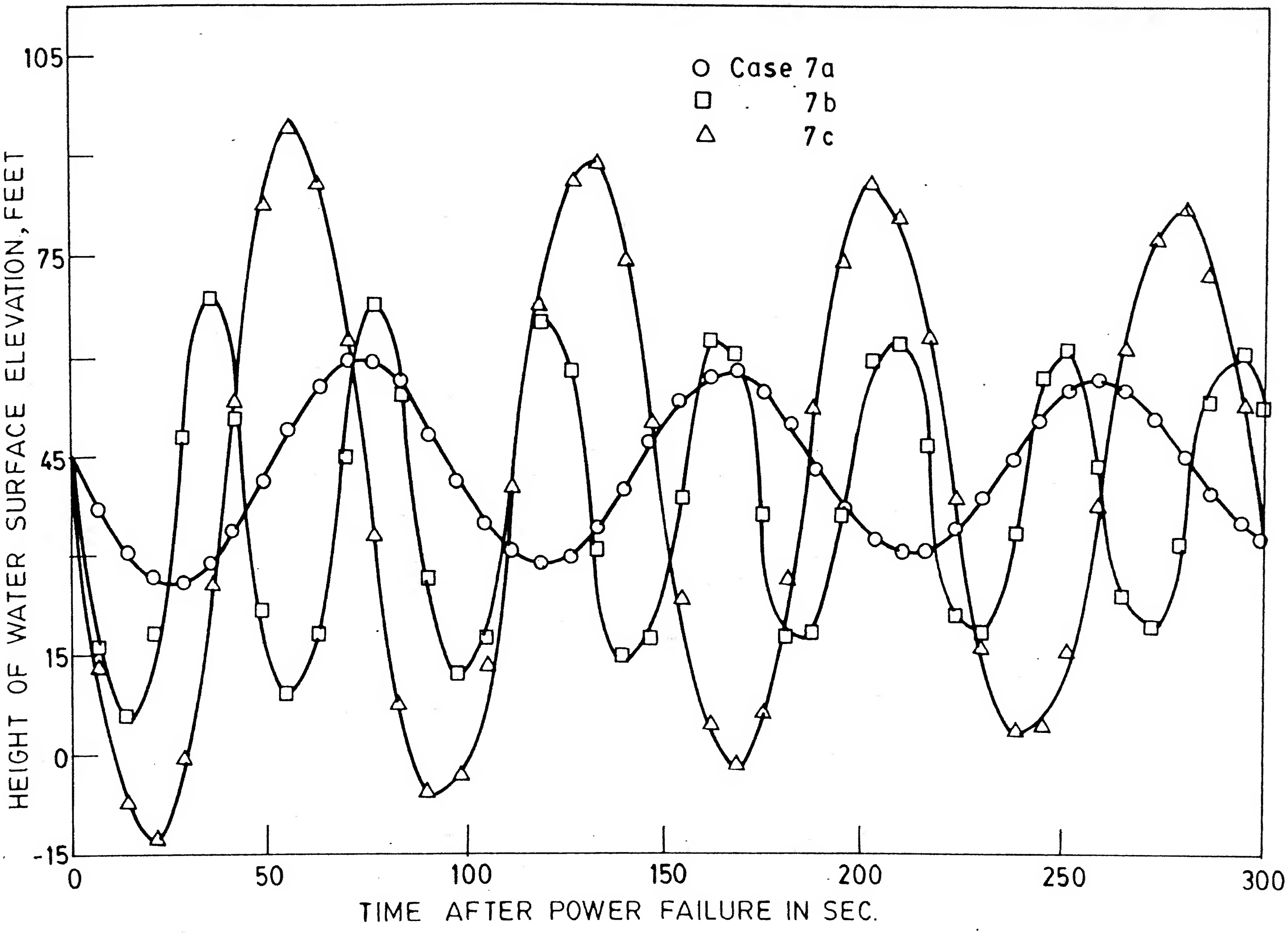

FIG. 4.3 WATER SURFACE ELEVATION CASE 7a, 7b AND 7c

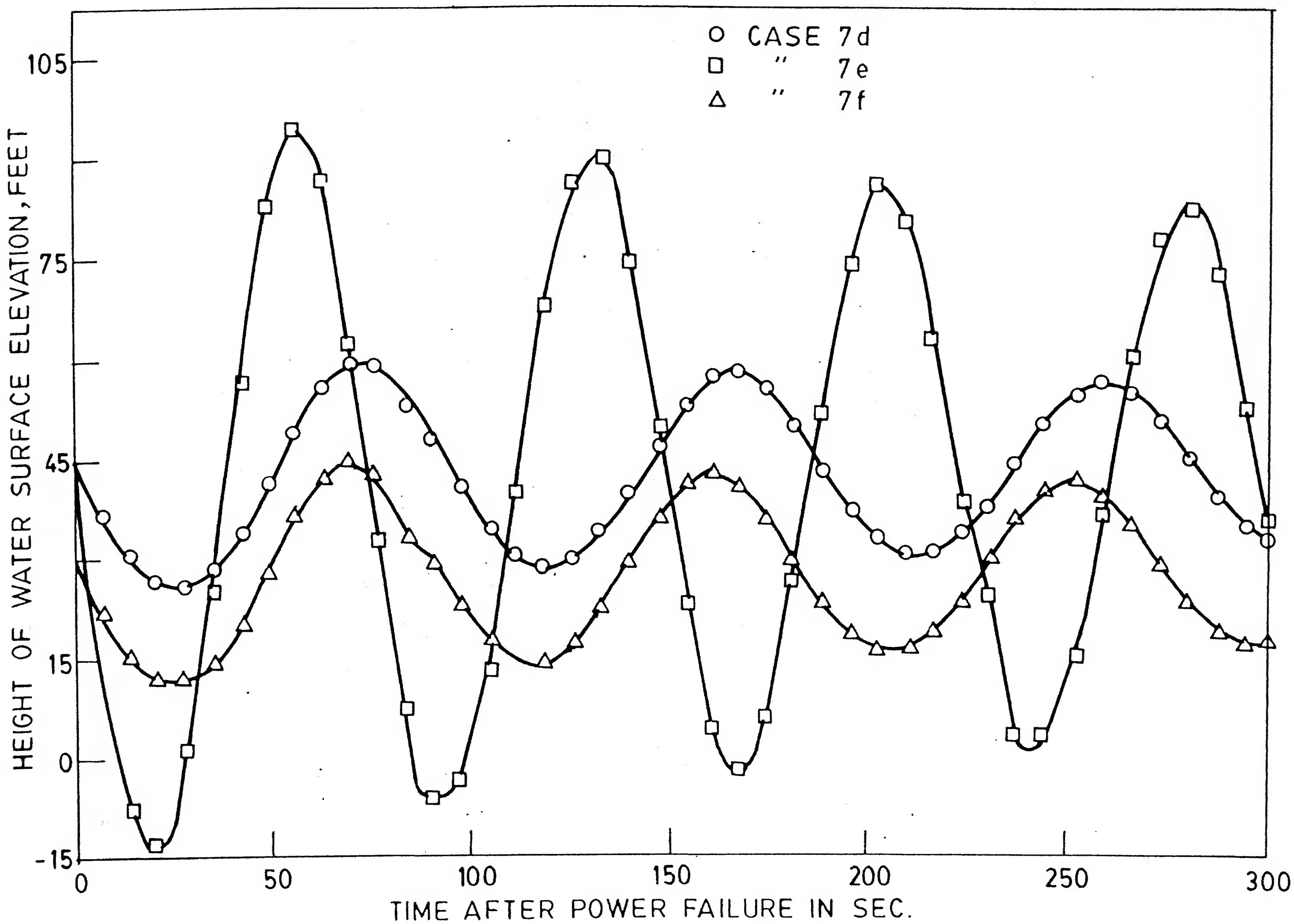

FIG. 4·4 WATER SURFACE ELEVATION FOR CASE 7d , 7e AND 7f

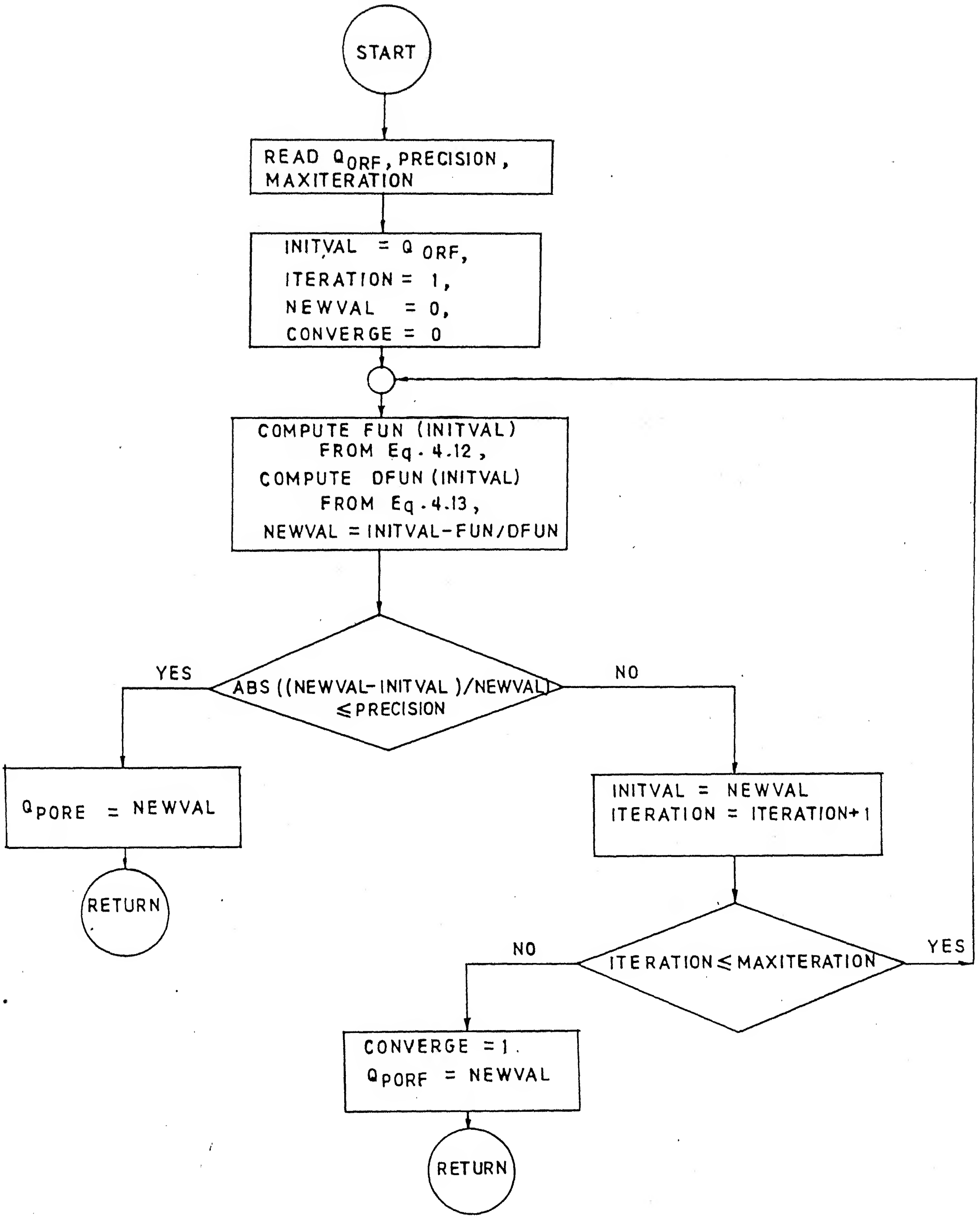

FIG. 4.5 NEWTON RAPHSON METHOD FOR COMPUTING DISCHARGE THROUGH ORIFICE OF AIR CHAMBER

TABLE 4.1, Basic Data And Results for different cases with Air Chamber

Item	Data& Results					
Case No	7a	7b	7c	7d	7e	7f
Coefficient of orifice for inflow	8.E-07	8.E-07	8.E-07	4.E-07	4.E-06	4.E-06
Coefficient of orifice for outflow	4.E-07	4.E-07	4.E-07	2.E-06	2.E-06	2.E-06
The value of constant (2.CO.a/QO.L)	50	10	50	50	50	50
Diameter of the Orifice	7.5	7.5	7.5	5.0	5.0	5.0
Water Surface Elevation from centre line of pipe in feet	45	45	45	45	45	45
Steady-State Air Chamber absolute Pressure head	186	186	186	186	186	186
Results						
Maximum water surface Elevation Elevation in feet	59.35	68.35	93.81	59.42	93.98	44.04
MMinimum water surface Elevation in feet	25.97	5.860	-13.08	25.86	-13.33	11.50
Elevation of the top of Air Chamber in feet	98.40	87.72	258.62	98.40	258.64	83.4
Maximum Air Chamber Pressure in feet	270.74	450.15	253.92	271.31	254.37	289.83
Minimum Air Chamber Pressure in feet	129.02	82.23	139.37	128.79	139.16	140.58

CHAPTER 5

SUMMARY, CONCLUSIONS AND SUGGESTIONS FOR FUTURE STUDY

5.1 Summary:

The Tracy Pumping Plant consists of six large units with each pair of pumps connected through a wye branch to a 15 ft. diameter, reinforced concrete pipe which extends approximately 1 mile to the upper canal. Each pump is driven by 22,500-hp motor and delivers water at the rate of 767 cu-ft. per sec. to the canal under a rated head of 197 ft. On the discharge side of each pump there is a butterfly valve which closes at a variable rate under the action of servomotor when power tripout occurs at the pump.

A comprehensive study of the hydraulic transients for parallel pump system for the Tracy Pumping Plant and for different modifications of the original plant involving a total of 14 different cases has been carried out. For this a computer program has been coded in FORTRAN- IV and run on Dec 1090 system at I.I.T. Kanpur. The system has been analysed without a valve, with a butterfly valve at the pump, a non-return valve at the pump, a combination of a non-return valve at the pump and an intermediate non-return valve at mid section of the discharge line. It has also been analysed with some suction length, increasing the length of discharge line, and doubling the moment of inertia. The system has also been analysed with an airchamber with a check valve upstream of the chamber, by

changing the different parameters involved in the computations. All the cases, being analysed under this thesis, for the transient problems, are due to sudden power tripout.

It has been found that maximum head is 1.4 times the rated head when, system has been analysed without valve. The system, analysed with butterfly valve has shown that 9 percent reducation in maximum head is achieved. The transients for non-return valve at the pump are so severe and quick that special precautions will have to be taken. In the case of resistance dominated system, the maximum head rise is equal to steady-state rated head, and column separation may occur with subsequent rejoining and development of high heads. In case of gravity loading,doubling of moment of inertia has decreased the maximum head and increased the minimum head, and so increase in moment of inertia will be benificial. The system studied with air chamber has indicated that important design parameter for the design of airchamber is $\frac{2CO.a}{QO.L}$, the values of upsurge and downsurge decrease with increase in this constant. The orifice diameter does not seem to have appreciable effect on upsurge and downsurge.

The findings of the present study provide valuable suggestions for effective and rational design of surge controlling equipments.

5.2 Conclusions:

The following conclusions emerge from the study:

a) Pressure surges in pump discharge lines subsequent to a power failure at the pump motors can be determined with satisfactory accuracy if the complete characteristics of the pump and control valves are known. Only the pump characteristics for the zone of normal operation were obtainable from the pump-performance data supplied by the manufacturer. This permitted an accurate determination of the water-hammer effects in the discharge line up to the point at which the flow reversed through the pump.

However, these data are sufficient for determining the possibility of water-column separation in the discharge line. Pump characteristics for abnormal operation and the characteristics of the butterfly valve are estimated for the remainder of the water-hammer solution.

b) Computation based on the method of characteristics are made to study the effect of intermediate non-return valves on surge pressures. Effect of valve location, and multiple valves are studied. One conclusion of practical interest is that there is no strong case for providing intermediate non-return valve.

c) Since the transients are so severe for complete failure with check valves for gravity loading, special precautions should be taken.

d) In case of gravity loading, doubling of moment of inertia decreases the maximum head and increases minimum head, so it can be concluded that increase in moment of inertia for gravity loading is benificial.

e) In case of resistance dominated loading maximum head developed is equal to steady-state head but the minimum pressure heads show that column-separation may occur with subsequent rejoining and development of high heads.

f) The important design parameter for the air vessel is $\frac{2CO.a}{QO.L}$, where CO is initial volume of air, if this parameter is increased, the values of upsurge and downsurge decrease, also the maximum air pressure decreases and the minimum air pressure increases.

g) The reduction in diameter of the air vessel causes increase in the upsurge and downsurge.

h) The change in the orifice diameter does not seem to have any effect on the surges for the small change that has been considered in the present study.

5.3 Suggestions for Future Study:

The following suggestions are made for further study on the basis of the present study:

a) Effect of a butterfly valve upstream from the air vessel with certain specified closure characteristics may be analysed.

b) Non-return valve with provision of bypass and its size effect may be analysed.

c) Non-return valve effects with delay in valve closure and random delays in the closure of valves may be studied.

d) The effect of pressure-regulating valves, and pressure-relief valves may be studied.

e) A comparative study of anairchamber and a surge tank in a pumping system may be carried out.

f) A program based on the same general concept as for parallel pump system maybe developed for series pump system and can be tested for more reliable and correct data.

REFERENCES

1. Allievi, K., '' Air Chamber for discharge lines'', Trans. ASME, Vol.59,Nov. 1937, pp 651-659.

2. Chaudhry M. Hanif. ''Applied Hydraulic Transients'' Van Nostrand Reinhold Company Book; New York 1979.

3. Evans, W.E., and Crawford, C.C., ''Design charts for Air chambers on pump lines,'' Trans. ASME. Vol.119, No. 2710,1954, pp. 1025-1045.

4. Knapp, R.T., ''Complete characteristics of centrifugal pumps and their use in prediction of Transient behaviour,'' Trans. ASME, Vol. 59, 1937.

5. Kinno.H, ''Water-hammer control in centrifugal pump systems,'' Journal of the Hydraulic Division, ASCE. Vol194, May 1968, pp.1143-1156.

6. Murthy, Sridharan and Prasad ''Effect of Intermediate non-return valves on Transients in Pumping Mains,'' Journal of Institution of Engineers (India)Vol.67, Part C13 November 1986.

7. Parmakian.J. '' Water-hammer Analysis,''Dover Publication Inc., New York, 1963, pp 78-81.

8. Parmakian,J, '' Pressure surges in Pump Installations,'' Trans. of ASCE Dec. 1953, paper No. 2760.

9. Parmakian.J, '' Pressure Surge Control at Tracy Pumping Pant,'' Proc. ASME Vol.59, Nov. 1937, pp 651-659.

10. Ross. L.M., '' San Francisco, Calif, '' Trans. ASME,'' S. Morgan Smith Company. New York- 1933.

11. Streeter, V.L., and Lai, Co., '' Water-hammer Analysis including fluid friction,'' Trans. ASCE, Vol.128, 1963, pp.1491-1524.

12. Streeter, V.L., and E. Benjamin Wylie, Hydraulic Transients, McGrow-Hill Book Co., New York, 1967.

APPENDIX

PROGRAM LISTING

```
------------------------------------------------------------------------
DEVENDRA       KUMAR      SINGH
[ NP = NUMBER    OF    PIPES IN    DISCHARGE    LINE         ]
[ NRLP = NUMBER   OF   REACHES ON LAST   PIPE                ]
[ NPP =  NUMBER   OF   PARALLE     PUMPS                     ]
[ QO   =  STEADY   STATE   DISCHARGE                         ]
[ NO   =  STEADY   STATE   PUMP   SPEED                      ]
[ AR   =  AREA   OF   THE   PIPE                             ]
[ A    =  WAVE   VELOCITY  IN THE   PIPE                     ]
[ FRI   =  FRICTION    FACTOR    FOR   THE    PIPE           ]
[ HRES   =    DOWNSTREAM   RESERVOIR    LEVEL                ]
[  D      =    DIAMETER   OF   THE   PIPE   USED             ]
[  L      =    LENGTH    OF   THE   DISCHARGE    PIPE        ]
[  FH   =    POINTS    ON     HEAD   CHARACTERISTICS   CURVE ]
[  FB    =    POINTS    ON    TORQUE  CHARACTERISTICS  CURVE ]
[  ER    =    PUMP   EFFICIENCY  OF THE PLANT                ]
[  NR    =    RATED  SPEED OF   THE   PUMP                   ]
[  HR    =    RATED  HEAD   OF    THE    PUMP                ]
[  QR    =    RATED    DISCHARGE    OF    THE    PUMP        ]
[ DTH =    THETHA  INTRVAL FOR STORING CHARACTERICTICS CURVE]
[ TLAST=TIME FOR WHICH TRANS. STATE CONDITIONS ARE TO BE    ]
[                COMPUTED                                   ]
[NPC=NO. OF POINTS ON CHARACTERISTICS  CURVE                ]
[DTH=THETA  INTERVAL FOR STORING CHARACTERISTIC CURVE       ]
------------------------------------------------------------------------
ANALYSIS OF TRANSIENTS IN A PIPELINE CAUSED BY PUMPS
------------------------------------------------------------------------
      REAL L,NR,NO,DQ
      DIMENSION Q(10,20),H(10,20),QP(10,20),HP(10,20)
     1,CA(10),F(10),CF(10),AR(10),A(10),L(10),N(10),
     2D(10),FH(80),FB(80),HMAX(10),HMIN(10),Y(60)
      COMMON /CP/ ALPHA,QR,V,CN,DALPHA,DV,BETA,C5,C6,NPP,T,CASE
      COMMON /PAR/ FH,FB,DTH,TAUO,TSTRT,HR,QO,HO,CV,HSUC
      COMMON /RAM/DXT,Y,C7,C8
      COMMON /RAMA/ HSTO,CO,HB,AM,C,CORF,AC,CAT,DT,ZL,QOR
------------------------------------------------------------------------
              GENERAL  DATA
------------------------------------------------------------------------
              DATA G/32.2/
------------------------------------------------------------------------
      READ(20,*) CASE
      WRITE(21,1001) CASE
01    FORMAT(18X,'CASE NO. OF TRASIENTS.=',F5.1/)
      IF (CASE.EQ.1.) GO TO 1000
      IF (CASE.EQ.3.) GO TO 1000
      IF (CASE.EQ.4.) GO TO 1000
      IF (CASE.EQ.5.) GO TO 1005
      IF (CASE.EQ.6.) GO TO 1005
      IF (CASE.EQ.7.) GO TO 1007
      IF (CASE.EQ.8.) GO TO 1007
------------------------------------------------------------------------
         DATA   FOR   VALVE   OPERATION   RELATIONSHIP
------------------------------------------------------------------------
      READ(20,*) DELH
```

```
      READ(20,*) MA,TV,DXT,TAUO,TAUF,(Y(I),I=1,MA)
      WRITE(21,27) MA,TV,DXT,(Y(I),I=1,MA)
7     FORMAT(8X,'NUMBER OF POINTS ON TAU VS TIME CURVE =',I2/8X,
     1'VALVE OPERATION TIME = ',F8.2,'SEC'/8X,
     2'TIME INTERVAL FOR STORING TAU  CURVE = ',F6.3,'SEC'/8X,
     3'STORED TAU VALUES'/8X,7F8.3/8X,9F8.3/)
--------------------------------------------------------------------
      READING AND WRITING INPUT DATA
--------------------------------------------------------------------
      IF (CASE.EQ.2.) GO TO 1000
005   READ(20,*) ARS,DS,LS
--------------------------------------------------------------------

      DATA    FOR   AIRCHAMBER    CALCULATIONS

--------------------------------------------------------------------
007   READ(20,*) HSTO,CO,HB,AM,AC,ZL,DAC,CORF,ORDIA,CONST,ARORF
      WRITE(31,1013) HSTO,CO,HB,AM,AC,ZL,DAC,CORF,ORDIA,CONST
013   FORMAT(8X,'STEADY STATE AIRCHAMBER ABSOLUTE PRESSURE HEAD =',F7.2
     1/8X,'STEADY-STATE AIR VOLUME ENCLOSED IN AIRCHAMBER =',F12.2
     2/8X,'BAROMETRIC PRESSURE HEAD =',F5.2
     3/8X,'EXPONENT IN THE POLYSTROPIC GAS EQUATION =',F5.1
     4/8X,'HORIZONTAL CROSS-SECIONAL AREA OF THE AIRCHAMBER =',F10.2
     5/8X,'HEIGHT OF THE LIQID SURFACE IN THE AIRCHAMBER ABOVE THE
     6 DATUM =',F7.2
     7/8X,'DIAMETER OF THE AIRCHAMBER =',F8.2
     8/8X,'COEFFICIENT OF ORRIFICE CORF =',E14.2
     9/8X,'DIAMETER  OF  THE  ORRIFICE =',F15.1
     10/8X,'THE VALUE OF CONSTANT (2.*CO*a/(QO*L)) =',F5.1/)

--------------------------------------------------------------------
000   READ(20,*) NP,NRLP,IPRINT,NPP,QO,NO,TLAST,HSUC,TSTRT
      WRITE(21,19) NP,NRLP,QO,NO,TLAST,NPP,HSUC
9     FORMAT(8X,'NUMBER OF PIPES =',I3/8X,'NUMBER OF REACHES ON LAST
     1PIPE=',I3/8X,'STEADY STATE DISCH. =',F8.3,'FEET3/S'2X,'STEADY
     2STATE PUMP SPEED =',F6.1,'RPM'/8X,'TIME FOR WHICH TRANS.
     3STATE COND.ARE TO BE COMPUTED =',F5.1,'SEC'/8X,'NUMBER OF
     4PARALLEL PUMPS =',I3/8X,'SUCTION SIDE RESERVOIR LEVEL=',F3.1/)
--------------------------------------------------------------------

      READING AND WRITING OF PUMP DATA

--------------------------------------------------------------------
      READ(20,*) NPC,DTH,QR,HR,NR,ER,WR2,(FH(I),I=1,NPC)
      READ(20,*) (FB(I),I=1,NPC)
      WRITE(21,23) NPC,DTH,QR,HR,NR,ER,WR2,(FH(I),I=1,NPC)
3     FORMAT(8X,'NUMBER OF POINTS ON CHARACTERISTIC CURVE =',I4/
     18X,'THETA INTRVAL FOR STORING CHARACTERISTIC CURVE =',
     2F4.0/8X,'RATED DISCH. =',F7.2,'FEET3/S'/8X,'RATED HEAD =',
     3F6.1,'FEET'/8X,'RATED PUMP SPEED =',F6.1,'RPM'/8X,'PUMP
     4EFFICIENCY =',F6.3/8X,'WR2 =',F10.2,'LB-FEET2'//8X,'POINTS ON
     5HEAD CHARAC'/(X,10F7.3))
      WRITE(21,25) (FB(I),I=1,NPC)
5     FORMAT(/8X,'POINTS ON TORQUE CHARACTERISTIC'/ (X,10F7.3))
--------------------------------------------------------------------
       DATA   FOR PIPES  VARIABLES
--------------------------------------------------------------------
      READ(20,*) (L(I),D(I),A(I),F(I),I=1,NP)
      WRITE(21,40)
      FORMAT(/8X,'PIPE NO',5X,'LENGTH',5X,'DIA',5X,'WAVE VEL.'
     1,5X,'FRIC FACTOR'/22X,'(FEET)'6X,'FEET',7X,'(FEET/S)' )
      WRITE(21,50) (I,L(I),D(I),A(I),F(I),I=1,NP)
      FORMAT(10X,I3,6X,F7.1,3X,F5.2,5X,F7.1,11X,F5.3)
```

```
      DT=L(NP)/(NRLP*A(NP))
      WRITE(21,51)
1     FORMAT(/8X,'PIPE NO',5X,'ADJUSTED WAVE VEL',26X,'(FEET/S)')
-----------------------------------------------------------------------
            CALCULATION OF PIPE CONSTANTS
-----------------------------------------------------------------------
      DO 60 I=1,NP
      AR(I) =0.7854*D(I)**2
      AUNADJ=A(I)
      AN=L(I)/(DT*A(I))
      N(I)=AN
      AN1=N(I)
      IF((AN-AN1).GE.0.5) N(I)=N(I)+1
      A(I)=L(I)/(DT*N(I))
      WRITE(21,55) I,A(I)
5     FORMAT(16X,I3,12X,F7.1)
      CA(I)=G*AR(I)/A(I)
      CAT=CA(I)
      CF(I)=F(I)*DT/(2.*D(I)*AR(I))
      F(I)=F(I)*L(I)/(2.*G*D(I)*N(I)*AR(I)**2)
0     CONTINUE
-----------------------------------------------------------------------
            COMPUTATION OF CONSTANTS FOR PUMPS
-----------------------------------------------------------------------
      THE FOLLOWING  CONSTANTS ARE FOR SI UNITS.FOR ENGLISH UNITS,
      REPLACE 93604.99 BY 595.875 AND 4.775 BY 153.744
-----------------------------------------------------------------------
      TR=(595.875*HR*QR)/(NR*ER)
      C5=CA(1)*HR
      C6=-(153.744*TR*DT)/(NR*WR2)
      ALPHA=NO/NR
      V=QO/(NPP*QR)
      DV=0.0
      DALPHA=0.0
      IF (CASE.LT.5.) GO TO 1006
      C7=NP*(CA(1)+CA(1))*QR/(CA(1)*CA(1)*HR)
      C15=0.013*LS/(ARS*ARS*DS*8.*G)
-----------------------------------------------------------------------
            CALCULATION OF STEADY STATE  CONDITIONS
-----------------------------------------------------------------------
      QOR1=-QO
      QOR=QOR1+36.54
      IF (CASE.EQ.8.) QOR1=0.
      IF (CASE.EQ.8.) QOR=0.0
      C=HSTO*CO**AM
006   IF (V.EQ.0.0) GO TO 65
      TH=ATAN2(ALPHA,V)
      TH = 57.296*TH
      GO TO 68
5     TH=0.0
8     CALL PARAB(TH,1,Z)
      HO=Z*HR*(ALPHA**2+V**2)
      H(1,1)=HO+HSUC
      HV = H(1,1)-DELH
      CALL  PARAB(TH,2,Z)
      BETA=Z*(ALPHA**2+V**2)
      DO 80  I=1,NP
      NN=N(I)+1
      DO 70 J=1,NN
      H(2,1)=H(1,6)
      H(I,J)=H(I,1)-(J-1)*F(I)*QO**2
      IF (I.NE.NP.AND.J.EQ.NN) H(I+1,1)=H(I,J)
      Q(I,J)=QO
      QV=QO/2.
```

```
      AOP=QO/2.
      AHS1=HSUC
      AHS2=AHS1-C15*QO*QO
      AHS3=AHS2-C15*QO*QO
      AHS4=AHS3-C15*QO*QO
      AHS5=AHS4-C15*QO*QO
0     CONTINUE
      HMAX(I)=H(I,1)
      H(1,1)=H(1,1)+DELH
      Ah=(H(1,1)-4.)/HR
      HMIN(I)=H(I,1)
0     CONTINUE
      NN=N(NP)+1
      HRES=H(NP,NN)
      T=0.0
      WRITE(21,85)
5     FORMAT(/3X,'TIME',2X,'ALPHA',4X,'V',4X,'PIPE',17X,
     1'HEAD(FEET)', /24X,'NO',5X,'(1)',6X,'(2)',4X,'(3)',5X,'(4)',5X,
     2'(5)',5X,'(6)',5X,'(N+1)',5X,'(Ah)',/)
      WRITE(24,401)
01    FORMAT(5X,'TIME',25X,'DISCH.FEET3/SEC)',/17X,'(1)',9X,'QV',7X,
     1'(3)',8X,'(4)',10X,'(5)',8X,'(6)',8X,'(N+1)')
      WRITE(22,403)
03    FORMAT(5X,'TIME',25X,'SUCTION HEAD (FEET)',/17X,'(AHS1)',
     14X,'(AHS2)',5X,'(AHS3)',4X,'(AHS4)',6X,'(AHS5)')
      WRITE(31,1016)
016   FORMAT(/9X,'TIME',5X,'ZL',13X,'CO',10X,'QOR',6X,'HSTO',/)
      WRITE(31,1018) T,ZL,CO,QOR1,HSTO
018   FORMAT(/5X,F7.1,3X,F7.2,3X,F12.2,3X,2F9.2)
0     K=0
      I=1
      NN=N(I)+1
      IF (CASE.EQ.4.) GO TO 1011
      WRITE(21,86) T,ALPHA,V,I,H(1,1),HV,H(1,2),H(1,3),H(1,4),
     1H(1,5),H(1,NN),Ah
      IF (CASE.NE.4.) GO TO 1015
011   WRITE(21,84) T,ALPHA,V,I,H(1,1),H(1,2),H(1,4),H(1,6),H(2,1),
     1H(2,4),H(2,6),AH
015   WRITE(24,400) T,AOP,QV,Q(1,2),Q(1,3),Q(1,4),Q(1,5),Q(1,NN)
00    FORMAT(2X,F7.1,2X,7F11.2)
      WRITE(22,402) T,AHS1,AHS2,AHS3,AHS4,AHS5
02    FORMAT(2X,F7.1,2X,5F11.2)
      WRITE(31,1017) T,ZL,CO,QOR,HSTO
      DO 89 I=2,NP
      NN=N(I)+1
6     FORMAT(3F7.2,2X,I2,2X,7F8.1,2X,F8.2)
4     FORMAT(F7.1,2F7.2,2X,I2,2X,7F8.1,2X,F8.2)
017   FORMAT(5X,F7.1,3X,F7.2,3X,F12.2,3X,2F9.2)
9     CONTINUE
50    T=T+DT
      K=K+1
      IF (T.GT.TLAST) GO TO 240
-------------------------------------------------------------------
           PUMP  AT  UPSTREAM   END
-------------------------------------------------------------------
      IF (CASE.EQ.7.) ALPHA=0.0
      IF (CASE.EQ.7.) V=0.0
      CN =Q(1,2)-H(1,2)*CA(1)-CF(1)*Q(1,2)*ABS(Q(1,2))
      IF (CASE.LT.7.) GO TO 1112
      CALL  GUIDE(ANS)
      IF (ANS.LE.0.0) CORF=CORF/2
      CALL GUIDE(ANS)
      IF (CASE.EQ.8.) CALL PUMP
      IF ((CASE.EQ.8.).AND.(V.LE.0.0)) V=0.0
```

```
      IF ((CASE.EQ.8.).AND.(V.LE.0.0)) ALPHA=0.0
      QP(1,1)=-ANS+NPP*V*QR
      ZP=ZL+0.5*(QOR+ANS)*DT/AC
      VP=CO-AC*(ZP-ZL)
      ZL=ZP
      QOR=ANS
      CO=VP
      IF (CASE.EQ.8.) GO TO 1113
      IF (CASE.EQ.7.) GO TO 1113
12    CALL PUMP
      IF ((CASE.EQ.3.).AND.(V.LE.0.0)) V=0.0
      IF ((CASE.EQ.3.).AND.(V.EQ.0.0)) ALPHA=0.0
      IF ((CASE.EQ.4.).AND.(V.LE.0.0)) V=0.0
      IF ((CASE.EQ.4.).AND.(V.EQ.0.0)) ALPHA=0.0
      QP(1,1)=NPP*V*QR
      QV=QP(1,1)/2.
      AQP=QV
      C8=CN/C5
113   HP(1,1)=(QP(1,1)-CN)/CA(1)
      HPOR=CORF*ANS*ABS(ANS)
      HPAR=HP(1,1)+HB-ZL-HPOR
      HSTO=HPAR
      IF (T.GE.TSTRT) HP(1,1)=(QP(1,1)-CN)/CA(1)-CV*QP(1,1)
     1*ABS(QP(1,1))-8.17
      IF (CASE.EQ.1.) CV=0.0
      IF (CASE.EQ.3.) CV=0.0
      IF (CASE.EQ.4.) CV=0.0
      IF (CASE.EQ.5.) CV=0.0
      IF (CASE.EQ.6.) CV=0.0
      IF (CASE.EQ.7.) CV=0.0
      IF (CASE.EQ.8.) CV=0.0
      HV=HP(1,1)-CV*QP(1,1)*ABS(QP(1,1))
      Ah=(HP(1,1)-4.)/HR
      IF (CASE.LT.5.)  GO TO 1003
      IF (CASE.EQ.7.) GO TO 1003
      AHS1=HSUC
      AHS2=AHS1-C15*QP(1,1)*ABS(QP(1,1))
      AHS3=AHS2-C15*QP(1,1)*ABS(QP(1,1))
      AHS4=AHS3-C15*QP(1,1)*ABS(QP(1,1))
      AHS5=AHS4-C15*QP(1,1)*ABS(QP(1,1))
C-----------------------------------------------------------------------
C            INTERIOR      POINTS
C-----------------------------------------------------------------------
003   DO 170 I=1,NP
        NN=N(I)
      DO 160 J=2,NN
      CN=Q(I,J+1)-CA(I)*H(I,J+1)-CF(I)*Q(I,J+1)*ABS(Q(I,J+1))
      CP=Q(I,J-1)+CA(I)*H(I,J-1)-CF(I)*Q(I,J-1)*ABS(Q(I,J-1))
      QP(I,J)=0.5*(CP+CN)
      C8=(CN+CP)/C5
      HP(I,J)=(CP-QP(I,J))/CA(I)-CV*QP(I,J)*ABS(QP(I,J))
      IF (T.GE.TSTRT) HP(I,J)=(CP-QP(I,J))/CA(I)
60      CONTINUE
70        CONTINUE
C-----------------------------------------------------------------------
C                    SERIES     JUNCTION
C-----------------------------------------------------------------------
      NP1=NP-1
      IF(NP.EQ.1) GO TO 178
      DO 175 I=1,NP1
      N1=N(I)
      NN=N(I)+1
      CN=Q(I+1,2)-CA(I+1)*H(I+1,2)-CF(I+1)*Q(I+1,2)*ABS(Q(I+1,2))
```

```
      CP=Q(I,N1)+CA(I)*H(I,N1)-CF(I)*Q(I,N1)*ABS(Q(I,N1))
      HP(I,NN)=(CP-CN)/(CA(I)+CA(I+1))
      HP(I+1,1)=HP(I,NN)
      QP(I,NN)=CP-CA(I)*HP(I,NN)
      QP(I+1,1)=CN+CA(I+1)*HP(I+1,1)
      IF ((CASE.EQ.4.).AND.(QP(I,NN).LE.0.0)) QP(I,NN)=0.0
      IF ((CASE.EQ.4.).AND.(QP(I+1,1).LE.0.0)) QP(I+1,1)=0.0
      IF ((CASE.EQ.4.).AND.(QP(I,NN).EQ.0.0)) HP(I,NN)=CP/CA(I)
      IF ((CASE.EQ.4.).AND.(QP(I+1,1).EQ.0.0)) HP(I+1,1)=-CN/CA(I+1)
75          CONTINUE
-------------------------------------------------------------------------
**        RESERVOIR  AT DOWNSTREAM   END
-------------------------------------------------------------------------
78    NN=N(NP)+1
      HP(NP,NN)=HRES
      CP=Q(NP,NN-1)+CA(NP)*H(NP,NN-1)-CF(NP)*Q(NP,NN-1)*
     1ABS(Q(NP,NN-1))
      QP(NP,NN)=CP-CA(NP)*HP(NP,NN)
      C8=CP/C5
**      STORING MAX. AND MIN. PRESSURES AND VARIABLES FOR NEXT TIME STEP
10    DO 230 I=1,NP
      NN=N(I)+1
      DO 220 J=1,NN
      Q(I,J)=QP(I,J)
      H(I,J)=HP(I,J)
      IF (H(I,1).GT.HMAX(I))  HMAX(I)=H(I,1)
      IF (H(I,1).LT.HMIN(I))  HMIN(I)=H(I,1)
20      CONTINUE
30         CONTINUE
      IF (K.EQ.IPRINT) GO TO 90
       GO TO  150
40    WRITE(21,250)
50    FORMAT(//10X,'PIPE NO.',5X,'MAX.PRESS.',5X,'MIN.PRESS.'/
     125X,'(FEET)',10X,'(FEET)'/)
      WRITE(21,260)    (I,HMAX(I),HMIN(I),I=1,NP)
60    FORMAT(12X,I3,7X,F7.1,9X,F7.1)
         STOP
         END
-------------------------------------------------------------------------
         SUBROUTINE    PUMP
-------------------------------------------------------------------------
      DIMENSION  FH(80),FB(80),Y(60)
      COMMON /CP/ ALPHA,QR,V,CN,DALPHA,DV,BETA,C5,C6,NPP,T,CASE
      COMMON  /PAR/ FH,FB,DTH,TAUO,TSTRT,HR,QO,HO,CV,HSUC
      COMMON /RAM/DXT,Y,C7,C8
      KK=0
      JJ=0
-------------------------------------------------------------------------
             COMPUTATION   OF  PUMP   DISCHARGE
-------------------------------------------------------------------------
      VE=V+DV
      ALPHAE=ALPHA+DALPHA
      JJ=JJ+1
0     IF (VE.EQ.0.0.AND.ALPHAE.EQ.0.0) GO TO 29
      TH=ATAN2(ALPHAE,VE)
      TH1=TH
      TH = TH*57.296
      IF (TH.LT.0.0) TH=TH+360.
      IF (TH1.LT.0.0)  TH1=TH1+6.28318
      GO TO 30
9     TH=0.0
      TH1=0.0
0     M=TH/DTH+1
      A1=FH(M)*M-FH(M+1)*(M-1)
```

```
      A2=(FH(M+1)-FH(M))/(DTH*0.017453)
      A3=FB(M)*M-FB(M+1)*(M-1)
      A4=(FB(M+1)-FB(M))/(DTH*0.017453)
      ALPSQ=ALPHAE*ALPHAE
      VESQ=VE*VE
      ALPV=ALPSQ+VESQ
-------------------------------------------------------------------------
      CALL PARAB1(T,TAU)
-------------------------------------------------------------------------
      IF (CASE.EQ.2.) GO TO 1004
      IF (CASE.EQ.1.) GO TO 1002
      IF (CASE.EQ.3.) GO TO 1002
      IF (CASE.EQ.4.) GO TO 1002
      IF (CASE.EQ.5.) GO TO 1002
      IF (CASE.EQ.6.) GO TO 1002
      IF (CASE.EQ.7.) GO TO 1008
002   CV=0.0
      C8=(CN*C5+CP*C5)/(C5*C5)
004   CV=8.17*2.*VE*VE*NPP*NPP*HR/(TAU*TAU*QO*QO*C5)
008   F1=C5*A1*ALPV+C5*A2*ALPV*TH1-QR*VE*NPP+CN+C5*HSUC/HR
      IF (CASE.EQ.7.) GO TO 1009
      IF (CASE.EQ.1.) F1=F1-C5*HSUC/HR
      IF (T.GE.TSTRT)
      1 F1=F1-C5*CV*QR*QR*VE*ABS(VE)*NPP*NPP/HR
009   F2=ALPHAE-C6*A3*ALPV-C6*A4*ALPV*TH1-ALPHA-C6*BETA
      F1AL=C5*(2.*A1*ALPHAE+A2*VE+2.*A2*ALPHAE*TH1)
      F1V=C5*(2.*A1*VE-A2*ALPHAE+2.*A2*VE*TH1)-QR*NPP
      IF (CASE.EQ.7.) GO TO 1010
      IF (T.GE.TSTRT)
      1 F1V=F1V-2*C5*CV*QR*QR*NPP*NPP*ABS(VE)/HR
      IF (CASE.EQ.5.) F1=A1*ALPV+A2*ALPV*TH1-C7*VE+C8
      IF (CASE.EQ.6.) F1=A1*ALPV+A2*ALPV*TH1-C7*VE+C8
      IF (CASE.EQ.5.) F1V=2.*A1*VE+2.*A2*VE*TH1-A2*ALPHAE-C7
      IF (CASE.EQ.6.) F1V=2.*A1*VE+2.*A2*VE*TH1-A2*ALPHAE-C7
      IF (CASE.EQ.5.) F1AL=2.*A1*ALPHAE+2.*A2*ALPHAE*TH1+VE*A2
      IF (CASE.EQ.6.) F1AL=2.*A1*ALPHAE+2.*A2*ALPHAE*TH1+VE*A2
010   F2AL=1.-C6*(2.*A3*ALPHAE+A4*VE+2.*A4*ALPHAE*TH1)
      F2V=C6*(-2.*A3*VE+A4*ALPHAE-2.*A4*VE*TH1)
      DENOM=F1AL*F2V-F1V*F2AL
      DALPHA=(F2*F1V-F1*F2V)/DENOM
      DV=(F1*F2AL-F2*F1AL)/DENOM
      ALPHAE=ALPHAE+DALPHA
      VE=VE+DV
      IF (ABS(DV).LE.0.001.OR.ABS(DALPHA).LE.0.001) GO TO 50
      IF (JJ.GT.30) GO TO 70
       GO TO 8
50    TH=ATAN(ALPHAE/VE)
0     TH=ATAN2(ALPHAE,VE)
      TH=57.296*TH
      IF (TH.LT.0.0) TH=TH+360.
        CALL   PARAB(TH,2,BETA)
      IF (MB.EQ.M)  GO TO 60
      MB=TH/DTH+1
      IF (MB.EQ.M) GO TO 60
      GO TO 8
0     DALPHA=ALPHAE-ALPHA
      DV=VE-V
      ALPHA=ALPHAE
      V=VE
      BETA =BETA*(ALPHA*ALPHA+V*V)
      RETURN
0     WRITE(21,80) T,ALPHAE,VE
0     FORMAT(8X,'*** ITERATIONS IN PUMP SUBROUTINE FAILED' /8X,
      1'T=',F8.2/8X,'ALPHAE =',F15.4/8X,'VE =',F6.3)
```

```
      STOP
         END
C-------------------------------------------------------------------
      SUBROUTINE  PARAB(X,J,Z)
C-------------------------------------------------------------------
      PARABOLA THRROUGH  THREE  POINTS ON CURVE
      (I) IS THE INTEGER PART OF THE FRACTION
      (R)  IS  THE  REAL  PART  OF  THE  FRACTION
      GENERALLY  VALUES OF (Y) ARE  KNOWN  FOR EQAL INCREMENT IN (X)
      IF VALUES OF (Y) ARE DESIRED FOR (X) WITH ORIGIN (0,0)
      ONE FIRST FINDS A PARABOLA THROUGH THE THREE POINTS WITH
      AXIS- VERTICAL BY TRANSFERING THE ORIGIN TO (X',Y')
      THERFORE Y' = A*X'**2+B*X'
      LET X(N+1)-X(N)=X(N)-X(N-1)=H
      THEN [Y(N+1)-Y(N)=A*H**2+B*H]
      AND  [Y(N-1)-Y(N)=A*H**2-B*H]
      A=[Y(N+1)+Y(N-1)-2.*Y(N)]/(2.*H**2)
      B=[Y(N+1)-Y(N-1)]/(2.*H**2)
      THEREFORE, [Y=[Y(N)+0.5*THETA**2((Y(N+1)+Y(N-1)-2.*Y(N))
               +THETA*((Y(N+1)-Y(N-1))*0.5]]
      IN  WHICH THETA*H  HAS BEEN SUBSTITUTED  FOR (X')

C-------------------------------------------------------------------
      COMMON /PAR/ FH,FB,DX,TAUO,TSTRT,HR,QO,HO,CV,HSUC
      DIMENSION FH(80),FB(80)
C-------------------------------------------------------------------
      I=X/DX
      R=(X-I*DX)/DX
      IF (I.EQ.0)R=R-1.
      I=I+1

      IF (I.LT.2) I=2
      GO TO (10,299 ),J
0     Z=FH(I)+0.5*R*(FH(I+1)-FH(I-1)+R*(FH(I+1)+FH(I-1)-2.*FH(I)))
      RETURN
99    Z=FB(I)+.5*R*(FB(I+1)-FB(I-1)+R*(FB(I+1)+FB(I-1)-2.*FB(I)))
      RETURN
      END
C-------------------------------------------------------------------
      SUBROUTINE PARAB1(T,TAU)
C-------------------------------------------------------------------
      COMMON /RAM/DXT,Y,C7,C8
      DIMENSION  Y(60)
C-------------------------------------------------------------------
      I=T/DXT
      R=(T-I*DXT)/DXT
      IF (I.EQ.0.0) R=R-1.
      I=I+1
      IF (I.LT.2) I=2
      TAU=Y(I)+0.5*R*(Y(I+1)-Y(I-1)+R*(Y(I+1)+Y(I-1)-2.*Y(I)))
      RETURN
      END
C-------------------------------------------------------------------
      THE FOLLOWING PROGRAM CALLS THE  NEWTON-RAPHSON SUBROUTINEP
C-------------------------------------------------------------------
      SUBROUTINE GUIDE(ANS)
      COMMON /RAMA/ HSTO,CO,HB,AM,C,CORF,AC,CAT,DT,ZL,QOR
      COMMON /CP/ ALPHA,QR,V,CN,DALPHA,DV,BETA,C5,C6,NPP,T,CASE
C-------------------------------------------------------------------
      EXTERNAL FUN,DFUN
      X01=0.0
      CALL ROOT(FUN,DFUN,X01,0.001,50,ID,ANS)
      RETURN
      END
```

```
-------------------------------------------------------------------------
      SUBROUTINE FOR FINDING THE ROOT OF AN EQUATION BY
-------------------------------------------------------------------------
      NEWTON-RAPHSON METHOD. FUN IS THE FUNCTION NAME.
      XINTL  IS  THE  INITIAL  TRIAL   ROOT.
      DFUN  IS THE DERIVATIVE OF THE FUNCTON  FUN.
      EPS IS THE PRESCRIBED ERROR LIMMIT OF THE ROOT.
      MAXIT IS THE MAXIMUMUM NUMBER OF ITERATIONS ALLOWED.
      INDIC IS AN INDICATOR SET TO 1 IF THERE IS  NO
      CONVERGENCE   IN  MAXIT    ITERATIONS.
        THE      ROOT   IS   RITURNED   IN  SOLN.
-------------------------------------------------------------------------
      SUBROUTINE ROOT(FUNCN,DFUNCN,XINTL,EPS,MAXIT,INDIC,SOLN)
-------------------------------------------------------------------------
      COMMON /RAMA/ HSTO,CO,HB,AM,C,CORF,AC,CAT,DT,ZL,QOR
      COMMON /CP/ ALPHA,QR,V,CN,DALPHA,DV,BETA,C5,C6,NPP,T,CASE
      INDIC=0
      DO 100 I=1,MAXIT
      FO=FUNCN(CN,XINTL,CAT,HB,ZL,QOR,DT,AC,CORF,CO,AM,C)
      DFO=DFUNCN(CN,XINTL,CAT,HB,ZL,QOR,DT,AC,CORF,CO,AM)
      XNEW=XINTL-(FO/DFO)
      IF (ABS((XNEW-XINTL)/XNEW) .LE. EPS) GO TO 120
      XINTL=XNEW
)0    CONTINUE
)1    INDIC=1
      SOLN=XNEW
      RETURN
0     SOLN=XNEW
      RETURN
      END
-------------------------------------------------------------------------
      THE DEFINITION OF THE  FUNCTIONS  ARE  GIVEN  BELOW
-------------------------------------------------------------------------
      FUNCTION FUN(CN,X,CAT,HB,ZL,QOR,DT,AC,CORF,CO,AM,C)
      FUN=((-CN-X)/CAT+HB-ZL-0.5*(X+QOR)*DT/AC-CORF*X*ABS(X))
     1*(CO-DT*0.5*(QOR+X))**AM-C
      RETURN
      END
      FUNCTION DFUN(CN,X,CAT,HB,ZL,QOR,DT,AC,CORF,CO,AM)
      DFUN=-((-CN-X)/CAT+HB-ZL-0.5*(X+QOR)*DT/AC-CORF*X*ABS(X))
     1*AM*DT*0.5*((CO-DT*0.5*(QOR+X)))**(AM-1)
     2+(-1/CAT-0.5*DT/AC-2.0*CORF*ABS(X))*((CO-DT*0.5*(QOR+X)))
     3**AM
      RETURN
14    END
-------------------------------------------------------------------------
                      THE   END
-------------------------------------------------------------------------
```